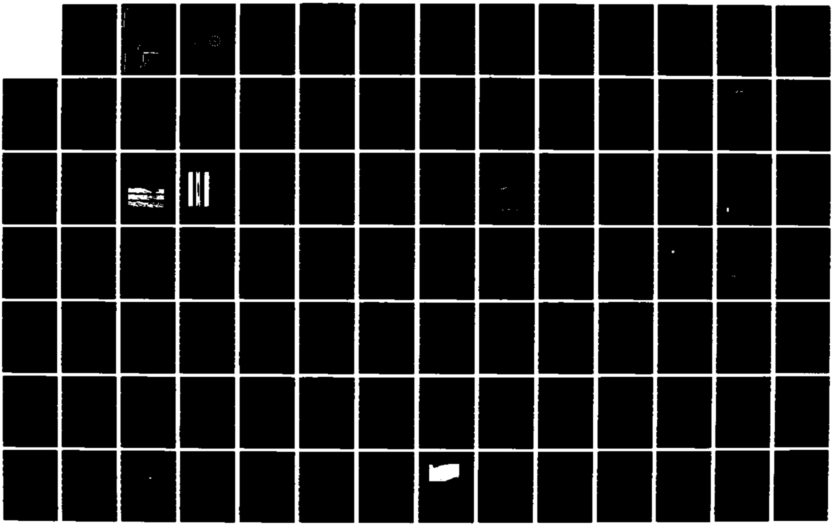
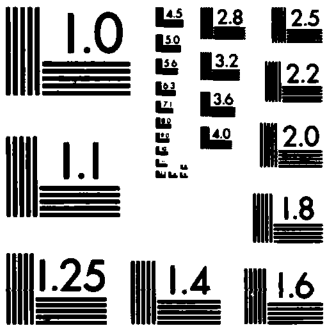
AD-A132 513 HANDBOOK OF OCEANOGRAPHIC WINCH WIRE AND CABLE TECHNOLOGY(U) RHODE ISLANDUUNIV KINGSTON GRADUATE SCHOOL OF OCEANOGRAPHY A H DRISCOLL 18 OCT 82 1/5

UNCLASSIFIED N00014-82-G-0116 F/G 8/10 NL





MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

(1)

HANDBOOK OF OCEANOGRAPHIC WINCH, WIRE AND CABLE TECHNOLOGY

EDITOR

ALAN H. DRISCOLL

HANDBOOK OF OCEANOGRAPHIC WINCH, WIRE AND CABLE TECHNOLOGY

PREPARED UNDER A GRANT FROM THE U.S. NAVY, OFFICE OF NAVAL RESEARCH AND THE NATIONAL SCIENCE FOUNDATION

EDITOR
ALAN H. DRISCOLL

OCTOBER 1982

Handbook of Oceanographic Winch, Wire, and Cable Technology

This Handbook has been prepared for the ship operator, engineer, scientist, or technician involved in the use and reliance upon the various deep sea winch, wires, and cables found in the oceanographic community. It brings together information that has been previously scattered among publications, catalogs, manufacturers literature, and unpublished data. Effort has been placed in the practical aspects of the winch and wire combinations while leaving the more specialized and detailed engineering aspects of the subject to individual investigation.

The various chapters in this Handbook have been prepared by recognized authorities in their specific fields. The editing of this volume has been restricted to providing continuity of material and has avoided altering the individual author style.

The Handbook had its beginning in late 1981 as a result of conversations with concerned ship users. At that time it became increasingly clear, through informal discussions, that some consolidated approach to the handling and understanding of both the deck machinery and wires used at sea by the oceanographic community would be required. Subsequent work performed by the University National Oceanographic Laboratory System (UNOLS), Technology Assessment Committee, further identified a need to concentrate upon developing a consistent understanding of winch and wire systems as they exist in the field.

This Handbook has been made possible through a joint effort by the Office of Naval Research and the National Science Foundation under grant No. N00014-82-G-0116 The project received enthusiastic support and encouragement from Mr. Keith Kaulum (ONR), Mr. Duane Tollaksen (ONR), Mr. Ronald LaCount (NSF), and Mr. Richard West (NSF).

Primary credit goes to the various chapter authors who have recognized a need for a work of this nature and who have donated their time and effort to make it a reality. Credit is also necessary to recognize the many commercial organizations who have donated both materials and time to the completion of the Handbook. Ms. Joanne May who typed the edited version and supervised the production of the camera ready copy provided an invaluable service; Mr. Carlos Correa and Mr. Paul Tattersall who produced the various graphics found in the text; Mrs. Doris Driscoll who proofread the manuscript; and Mrs. Anne Barrington who produced the final copy deserve much credit and thanks.

Alan H. Driscoll, Ed.

Albert G. Berian

The Rochester Corporation
Electro-Mechanical Cables (2)

Henri O. Berteaux, Research Specialist

Woods Hole Oceanographic Institution
Equipment Design (7)

Raymond J. Boudreaux, Jr., P.E., President

Victory Engineering Services
Winch Hydraulic Systems (11)

Alan H. Driscoll, Executive Director

Graduate School of Oceanography, University of Rhode Island
Winch and Wire Documentaiton and Instrumentation (5) (12)

Philip T. Gibson, P.E., President

Tension Member Technology
Rope and Cable Operational Characteristics (4)

Emile E. Grignard, President

Grignard Chemical Company, Inc.
Wire Rope and Cable Lubrication (3)

H. Wayne Johnson

Smatco Deck Division
Double Drum Traction Winch (9)

John M. LeDet, President

Smatco Deck Division
Double Drum Traction Winch (9)

W. A. Lucht, Technical Director

U.S. Steel, Wire Rope Products Division
3 x 19 Wire Rope (1)

Michael J. Markey, P.E., Engineering

Markey Machinery Company, Inc.
Single Drum Winches (8)

C. Robert Shaw, Chief Engineer

Union Metal Manufacturing Company, Electroline Division
Rope and Cable Terminations (6)

Reynolds E. Smith, Vice President Engineering

Lebus International, Inc.
Winch Level Winding (10)

Traditionally winches, wires, and their associated sheaves have always been treated as a loose collection of individual components which are supposed to perform as we expect them to. This approach was a relatively safe one when the instrument loads being fielded were light and the science less complex. In today's world, however, instrument loads have increased, the size and complexity of instruments has grown, and the competition for research dollars has forced the sea-going scientist to attempt to maximize his available vessle time.

All of these factors are placing additional stresses on the research vessels' deck machinery and wire combinations to the point where mechanical breakdown, wire failure, and equipment loss are becoming all too frequent. Because of this, we feel it is necessary to take a detailed look at our winches, wires and cables with an eye toward improving vessel efficiency and extending the useful working life of the ropes involved in deep ocean research. We should never lose sight of the fact that the deck machinery and its associated ropes exist only for the direct support of the user scientist and that it is the responsibility of the ship operating personnel, staff engineer, and technicians to ensure that a reliable system is available to that user scientist on demand.

The word "system" has been used to describe a research vessel's winches, wires, and sheave train because it is no longer realistic to consider each of these items as individual and separate entities. They are, instead, components of a dynamically complex system where the individual parts have a definite relationship to one another. It is this relationship which will be explored in this Handbook.

One word of caution when dealing with the evaluation of winch and wire systems, and that is the fact that achieving a calculated ideal operating condition is not always possible given the physical constraints imposed by research vessel size. When a departure is made from the ideal, the entire system is affected and an analysis of its new capabilities should be performed in order to establish its operating limits. Additionally, problems which occur in existing winch and wire systems are rarely a single point problem, but instead, are a function of several component problems. When problems arise, the entire system should be evaluated and appropriate corrections made.

This Handbook contains adequate information to initiate a careful review of existing winch and wire systems and to provide data pertinent to the upgrading of these systems. In addition, the material contained in this Handbook will be found useful where total winch replacements are planned. It is the hope of the authors that this Handbook will serve as a ready reference to the oceanogrpahic community now and in the future.

TABLE OF CONTENTS

PREFACE i

CONTRIBUTORS ii

INTRODUCTION iv

CHAPTER ONE - 3 X 19 OCEANOGRAPHIC WIRE ROPE 1-1

1.0 CONSTRUCTION AND MATERIALS 1-2

2.0 TESTING 1-2

2.1 BREAKING LOAD 1-2
2.2 0.2% OFFSET YIELD 1-4
2.3 ROTATION 1-4
2.4 MODULUS OF ELASTICITY 1-4

3.0 PROTECTIVE COATINGS 1-4

3.1 GALVANIZE 1-4
3.2 PLASTIC JACKETED WIRE ROPE 1-5
3.3 LUBRICATION 1-5

4.0 STRESS RELIEVING WIRE ROPE 1-5

5.0 SPOOLING AND STORAGE 1-6

5.1 LEBUS LAGGING 1-6
5.2 SMOOTH FACED DRUM 1-6

6.0 INSTALLATION OF TORQUE BALANCED ROPE 1-10

7.0 STAINLESS STEEL TORQUE BALANCED ROPE 1-12

8.0 ENDLESS NASH-TUCK SPLICE 1-15

9.0 WIRE ROPE RETIREMENT CRITERION 1-20

10.0 MINIMUM SHEAVE TREAD DIAMETERS 1-21

11.0 SHEAVE GROOVES 1-21

12.0 FLEET ANGLES 1-23

13.0 WIRE ROPE SPECIFICATIONS 1-24

14.0 NON-DESTRUCTIVE TESTING 1-25

14.1 INDIVIDUAL AC/DC UNITS 1-25
14.2 UNITIZED AC/DC UNITS 1-30

REFERENCES 1-32

CHAPTER TWO -
OCEANOGRAPHIC ELECTRO-MECHANICAL CABLES 2-1

1.0 CONSTRUCTION CHARACTERISTICS 2-4

1.1 COINCIDENCE 2-4
1.2 CENTER STRENGTH MEMBER 2-4
1.3 BRAIDED OUTER STRENGTH MEMBER 2-5
1.4 ELECTRO MECHANICAL WIRE ROPE 2-5
1.5 OUTER SINGLE SERVED STRENGTH MEMBER 2-6
1.6 OUTER DOUBLE SERVED STRENGTH MEMBER 2-6
1.7 3-4-5 LAYER SERVED STRENGTH MEMBER 2-7

2.0 WORKING ENVIRONMENT 2-7

2.1 FLEXING 2-7
2.2 ABRASION 2-7
2.3 TENSION CYCLING 2-8
2.4 CORROSION 2-8
2.5 FISH BITE 2-9
2.6 ABRASION RATE FACTORS 2-9
2.7 KINKING 2-9
2.8 CRUSHING 2-9

3.0 PARTS OF CONTRA-HELICALLY ARMORED E-M CABLE ... 2-9

3.1 DIRECTION OF LAY 2-10
3.2 LAY ANGLE 2-10
3.3 PREFORM 2-11
3.4 HEIGHT OF HELIX 2-11
3.5 PERCENT PREFORM 2-12
3.6 LENGTH OF LAY 2-12
3.7 PITCH DIAMETER 2-12
3.8 NUMBER OF ARMOR WIRES 2-13
3.9 ARMOR COVERAGE 2-14
3.10 SQUEEZE 2-14
3.11 CORE 2-15
3.12 WATER BLOCKED CORE 2-15

4.0 PERFORMANCE CHARACTERISTICS
OF C-H-A, E-M CABLES 2-16

4.1 TORQUE BALANCE 2-16
4.2 TWIST BALANCE 2-17
4.3 CRUSH RESISTANCE 2-17
4.4 CORROSION RESISTANCE 2-18
4.5 ABRASION RESISTANCE 2-19
4.6 ELONGATION 2-20
4.7 SEA WATER BUOYANCY 2-20
4.8 BREAKING STRENGTH 2-20

5.0 MANUFACTURING PROCESSES FOR E-M CABLES 2-22

5.1 CONDUCTOR STRANDING 2-22
5.2 INSULATION 2-22
5.3 WET TEST 2-22
5.4 CABLING 2-23
5.5 BRAIDING 2-23
5.6 SERVING 2-23
5.7 JACKETING 2-23
5.8 ARMORING 2-24
5.9 PRESTRESSING 2-24

6.0 HANDLING E-M CABLES 2-27

6.1 STORAGE BEFORE USE 2-27
6.2 SPOOLING EFFECT ON E-M CABLES 2-28
6.3 SMOOTH DRUM SPOOLING 2-29
6.4 TENSION SPOOLING OBJECTIVES 2-29
6.5 TENSION FOR SPOOLING 2-29
6.6 LOWER SPOOLING TENSIONS 2-30
6.7 GROOVED DRUM SLEEVES 2-30
6.8 SHEAVES 2-30

7.0 FIELD INSPECTION AND TESTING 2-31

7.1 GENERAL 2-31
7.2 REQUIRED INSPECTIONS 2-34
7.3 CABLE RECORD BOOK 2-34
7.4 CABLE LOG 2-34
7.5 INSPECTION 2-34
7.6 VISUAL INSPECTION PRACTICES 2-35
7.7 ARMOR TIGHTNESS INSPECTION 2-35
7.8 LAY LENGTH OF THE OUTER ARMOR 2-37
7.9 CONDUCTOR ELECTRICAL RESISTANCE 2-39
7.10 OUTSIDE DIAMETER 2-39
7.11 NEED FOR LUBRICATION 2-40
7.12 LOCATION OF OPEN CONDUCTOR 2-43
7.13 FAULT LOCATION 2-44
7.14 RE-REELING 2-44
7.15 CABLE LENGTH DETERMINATION 2-45

8.0 RETIREMENT CRITERIA 2-45

8.1 CONSIDERATIONS 2-45
8.2 BROKEN WIRE CRITERIA 2-46
8.3 LIFE CYCLE CRITERIA 2-47
8.4 NON-DESTRUCTIVE TESTING 2-48

9.0 CABLE MATERIALS 2-48

9.1 CONDUCTORS 2-48
9.2 INSULATIONS 2-50
9.3 SHIELDING 2-50
9.4 JACKETS 2-51
9.5 ARMOR 2-51

10.0 CONTRA-HELICALLY ARMORED E-M CABLE SPECS 2-53

10.1 PERFORMANCE vs. CONSTRUCTION SPECIFICATION 2-53
10.2 CONSTRUCTION SPECIFICATION 2-53
10.3 PERFORMANCE SPECIFICATION 2-54

11.0 AVAILABLE CABLE SERVICES 2-55

11.1 GENERAL 2-55
11.2 SPOOLING 2-57
11.3 SPLICING 2-57
11.4 FAULT LOCATION 2-57
11.5 RECONDITIONING 2-57
11.6 MAGNETIC MARKING 2-58

12.0 APPENDICIES 2-61

13.0 BIBLIOGRAPHY 2-94

CHAPTER THREE -
WIRE ROPE AND E-M CABLE LUBRICATION 3-1

1.0 INTRODUCTION 3-2

2.0 WIRE LUBRICATION THEORY 3-2

2.1 WIRE RE-LUBRICATION 3-2
2.2 CORROSION PROTECTION 3-3
2.3 SPLASH ZONE CORROSION 3-4
2.4 MARINE ATMOSPHERIC CORROSION 3-5
2.5 CORROSION DURING WIRE IMMERSION 3-5

3.0 WIRE LUBRICANTS AND CORROSION PROTECTION 3-6

3.1 INCORPORATION OF RUST
PREVENTATIVE DURING MANUFACTURE 3-8
3.2 TESTING 3-8
3.3 RUST PREVENTATIVES 3-10

4.0 COMPATIBILITY OF LUBRICANT
AND RUST PREVENTATIVE 3-11

4.1 SELECTION OF A FIELD DRESSING 3-11
4.2 [illegible]VANIZING OF ROPES AND CABLES 3-12

5.0 FIELD APPLICATION OF LUBRICANTS
AND RUST PREVENTATIVES 3-12

6.0 FIELD APPLICATION TECHNIQUES 3-13

6.1 CABLE DRYING 3-13
6.2 FIELD DRESSING APPLICATION 3-14

7.0 WIRE PRESERVATION DURING LONG TERM STORAGE 3-16

REFERENCES 3-17

CHAPTER FOUR -
WIRE ROPE AND CABLE OPERATIONAL CHARACTERISTICS 4-1

1.0 INTRODUCTION 4-2

2.0 REACTION OF WIRE ROPES
AND CABLES TO TENSILE LOADING 4-2

2.1 SIGNIFICANT TENSION INDUCED STRESSES 4-2
2.2 WIRE ROPE AND CABLE TORQUE 4-3
2.3 CABLE HOCKLING AND KINKING 4-5

3.0 REACTION OF A WIRE ROPE OR CABLE TO BENDING ... 4-7

3.1 WIRE BENDING STRESSES 4-7
3.2 RELATIVE MOTIONS AMONG
ROPE AND CABLE COMPONENTS 4-8
3.3 STRENGTH REDUCTION DUE TO BENDING 4-9
3.4 OTHER CONSIDERATIONS RELATIVE
TO ROPE AND CABLE BENDING 4-10

4.0 ROPE AND CABLE INTERACTION
WITH SYSTEMS COMPONENTS 4-14

4.1 WINDING ON DRUMS 4-14
4.2 SHEAVES 4-15
4.3 CABLE REEVING CONFIGURATIONS 4-18

5.0 WIRE ROPE AND CABLE FATIGUE DATA 4-19

6.0 ACCEPTABLE TESTS OF WIRE ROPES AND CABLES 4-22

6.1 WIRE ROPES 4-22
6.2 ELECTRO-MECHANICAL CABLES 4-23

CHAPTER FIVE - WIRE AND WINCH DOCUMENTATION 5-1

1.0 INTRODUCTION 5-2

2.0 WIRE AND CABLE DOCUMENTATION 5-3

2.1 WIRE AND CABLE IDENTIFICATION 5-3
2.2 WIRE SPECIFICATIONS 5-4
2.3 WIRE DOCUMENTATION SYSTEM 5-4

3.0 WIRE USE RECORD 5-7

4.0 WIRE INVENTORY RECORD 5-9

5.0 SHORE BASE WIRE LOG 5-11

6.0 WIRE FAILURE REPORT 5-12

6.1 SHIPBOARD ANALYSIS 5-15
6.2 ENGINEERING ANALYSIS 5-17
6.3 CORRECTIVE ACTION 5-17

7.0 WINCH DOCUMENTATION 5-20

8.0 SUMMARY 5-22

REFERENCES 5-23

CHAPTER SIX - ROPE AND CABLE TERMINATIONS 6-1

1.0 INTRODUCTION 6-2

2.0 SELECTION CRITERION 6-3

2.1 TYPE AND SIZE OF WIRE OR CABLE 6-3
2.2 CORROSION POTENTIAL 6-3
2.3 LOADING AND CABLE FATIGUE 6-4
2.4 TERMINAL EFFICIENCY 6-4
2.5 ASSEMBLY REQUIREMENTS AND COST 6-4

3.0 INSTALLATION PROCEDURES 6-5

3.1 WIRE ROPE CLIPS 6-5
3.2 WIRE ROPE THIMBLES 6-9
3.3 OPEN WEDGE TERMINATIONS 6-10
3.4 POURED ON SPELTER SOCKETS 6-11
3.5 COMPRESSED SLEEVES (NICOPRESS) 6-19
3.6 SWAGED TERMINATIONS 6-21
3.7 MECHANICAL TERMINATIONS
(ELECTROLINE OR FIEGE) 6-22

4.0 ELECTRO-MECHANICAL TERMINATIONS 6-28

4.1 ELECTROLINE E-M CABLE TERMINATIONS 6-28
4.2 INSTALLATION PROCEDURES 6-29
4.3 COMBINATION MECHANICAL/EPOXY TERMINATIONS 6-33
4.4 HELICALLY WOUND TERMINATIONS 6-37

CHAPTER SEVEN - EQUIPMENT LOWERING MECHANICS 7-1

1.0 INTRODUCTION 7-2

2.0 MECHANISMS CAUSING LOWERING CABLE DAMAGE 7-2

2.1 STATIC AND QUASI-STATIC TENSILE LOADS 7-3
2.2 WIRE INDUCED DYNAMIC LOADS 7-4
2.3 ZERO LOAD 7-6
2.4 SNAP LOADS 7-7

3.0 PREDICTING CABLE LOADS 7-8

3.1 IMMERSED WEIGHT STATIC LOAD 7-8
3.2 HYDRODYNAMIC RESISTANCE QUASI-STATIC LOAD 7-8
3.3 TERMINAL VELOCITY-ZERO LOAD 7-12
3.4 VIRTUAL MASS-INERTIA LOADS 7-13
3.5 ALL FORCES CONSIDERED-
STEADY STATE PEAK TENSIONS 7-15
3.6 SNAP LOADS 7-19
3.7 ADVANCED CABLE DYNAMICS 7-23

4.0 RECOMMENDATIONS 7-25

4.1 EQUIPMENT DESIGN CONSIDERATIONS 7-25
4.2 EQUIPMENT HANDLING CONSIDERATIONS 7-27

REFERENCES 7-29

BIBLIOGRAPHY 7-30

CHAPTER EIGHT - SINGLE DRUM WINCH DESIGN 8-1
1.0 THEORY OF OPERATION 8-2
2.0 WINCH CAPACITIES 8-7
2.1 DRUM CHARTS 8-7
2.2 DRAG FORCES 8-11
2.3 SPECIAL APPLICATIONS 8-13
2.4 GEAR RANGES 8-13
3.0 WINCH POWER OPTIONS 8-15
3.1 HYDRAULIC WINCH DRIVES 8-16
3.2 ELECTRIC WINCH DRIVES 8-18
4.0 WINCH CONTROLS 8-19
5.0 GROOVED SHELLS AND FAIRLEAD DRIVES 8-20
6.0 INSTRUMENTATION 8-22
7.0 CONSTRUCTION DETAILS 8-24
8.0 ORDERING INFORMATION 8-26
CHAPTER NINE - DOUBLE DRUM TRACTION WINCH DESIGN 9-1
1.0 THEORY OF OPERATION 9-2
2.0 WINCH POWER vs. TORQUE 9-6
2.1 A.C. ELECTRIC DRIVE SYSTEM 9-7
2.2 D.C. ELECTRIC DRIVE SYSTEM 9-8
2.3 ELECTRO-HYDRAULIC DRIVE SYSTEM 9-8
2.4 DIESEL-HYDRAULIC DRIVE SYSTEM 9-11
3.0 WINCH SIZING TO USER NEED 9-11
4.0 DRUM AND GROOVE DESIGN 9-13
5.0 MECHANICAL EFFICIENCY OF DOUBLE DRUM 9-19
REFERENCES 9-27
CHAPTER TEN -
ROPE AND CABLE LEVELWINDING AT THE WINCH 10-1
1.0 INTRODUCTION 10-2
2.0 THE LEBUS SHELL 10-3
2.1 GROOVING PATTERN 10-3
2.2 SOFT CROSSOVER SPOOLING 10-6
3.0 DESIGN CONSIDERATIONS 10-7
3.1 CABLE DIAMETER 10-7
3.2 DRUM CONSTRUCTION 10-7
3.3 LEBUS SHELL MATERIALS 10-11
3.4 WINCH DRUM FILLET WELD 10-11
3.5 DRUM CORE SIZE 10-12
3.6 HELIX CONSIDERATIONS 10-12

4.0 LEVEL WIND CONSIDERATIONS 10-12

4.1 LEVEL WIND TYPES 10-14
4.2 DIAMOND SCREW DRIVES 10-14
4.3 AUTOMATIC FLEET ANGLE COMPENSATOR 10-16
4.4 SINGLE LAYER SPOOLING 10-18

5.0 SPECIFICATIONS FOR GROOVED SHELLS AND COMPENSATORS 10-18

5.1 GROOVED SHELL 10-18
5.2 FLEET ANGLE COMPENSATOR 10-20

CHAPTER ELEVEN - WINCH HYDRAULIC SYSTEMS 11-1

1.0 INTRODUCTION 11-3

2.0 DESIGN AND OPERATION OF HYDRAULIC SYSTEMS 11-3

2.1 FUNDAMENTALS OF HYDRAULIC SYSTEMS 11-3
2.2 OPEN LOOP CIRCUITS 11-5
2.3 CLOSED LOOP "TRANSMISSION" CIRCUIT 11-5

3.0 HYDRAULIC CIRCUITS 11-7

3.1 INSTALLATION AND PLUMBING 11-8
3.2 COMPLEMENTARY COMPONENTS 11-8

4.0 HYDRAULIC FLUIDS (PETROLEUM & FIRE-RESISTANT TYPES) 11-10

4.1 FLUIDS VOCABULARY 11-13
4.2 PETROLEUM FLUIDS 11-14
4.3 FIRE-RESISTANT FLUIDS 11-14
4.4 SYNTHETIC FLUIDS 11-14
4.5 WATER GLYCOL FLUIDS 11-15
4.6 ASTM VISCOSITY CLASSIFICATION FOR INDUSTRIAL SYSTEMS 11-15

5.0 CLEANLINESS OF HYDRAULIC SYSTEMS 11-17

5.1 STORAGE AND HANDLING 11-17
5.2 FLUIDS IN SERVICE 11-17
5.3 GRAVITY SETTLING 11-18
5.4 CENTRIFUGING 11-19
5.5 CONTINUOUS RECONDITIONING 11-19
5.6 FILTRATION FOR FLUID POWER SYSTEMS 11-20
5.7 FILTER SELECTION FACTORS 11-21

6.0 HYDRAULIC SYSTEM MAINTENANCE 11-23

6.1 DRAIN SCHEDULES 11-24
6.2 CLEANING AND FLUSHING 11-25

7.0 TROUBLE SHOOTING HYDRAULIC SYSTEMS 11-29

7.1 IMPROPER OPERATION OF PUMP 11-29

7.2 IMPROPER OPERATION OF ACTUATING MECHANISM 11-33

7.3 IMPROPER OPERATION OF ACCUMULATOR 11-34

7.4 EXCESSIVE HEATING OF OIL IN SYSTEM 11-35

7.5 IMPROPER OPERATION OF FLUID MOTORS 11-36

CHAPTER TWELVE and - USEFUL INFORMATION................. 12-1

INDEX

CHAPTER I

3X19 Oceanographic Wire Rope

W. A. LUCHT

1.0 CONSTRUCTION AND MATERIALS	1-2
2.0 TESTING	1-2
2.1 BREAKING LOAD	1-2
2.2 0.2% OFFSET YIELD	1-4
2.3 ROTATION	1-4
2.4 MODULUS OF ELASTICITY	1-4
3.0 PROTECTIVE COATINGS	1-4
3.1 GALVANIZE	1-4
3.2 PLASTIC JACKETED WIRE ROPE	1-5
3.3 LUBRICATION	1-5
4.0 STRESS RELIEVING WIRE ROPE	1-5
5.0 SPOOLING AND STORAGE	1-6
5.1 LEBUS LAGGING	1-6
5.2 SMOOTH FACED DRUM	1-6
6.0 INSTALLATION OF TORQUE BALANCED ROPE	1-10
7.0 STAINLESS STEEL TORQUE BALANCED ROPE	1-12
8.0 ENDLESS NASH-TUCK SPLICE	1-15
9.0 WIRE ROPE RETIREMENT CRITERION	1-20
10.0 MINIMUM SHEAVE TREAD DIAMETERS	1-21
11.0 SHEAVE GROOVES	1-21
12.0 FLEET ANGLES	1-23
13.0 WIRE ROPE SPECIFICATIONS	1-24
14.0 NON-DESTRUCTIVE TESTING	1-25
14.1 INDIVIDUAL AC/DC UNITS	1-25
14.2 UNITIZED AC/DC UNITS	1-30
REFERENCES	1-32

1.0 CONSTRUCTION AND MATERIALS

3 x 19 torque balanced oceanographic wire rope is a relatively new invention in the wire rope industry. It was first put to use as a winch line at Woods Hole with Scripps following a very short time later.

This was during the period a tapered 6-strand wire rope was being tried for use in deep water. The tapered 6-strand rope, as well as all other 6-strand, non-torque balanced rope, unwind when suspending a free hanging load. Any momentary sudden release of tension permits the wire rope to start winding back to its original design because of its spring-like properties. In so doing, hockles and kinks are formed, which are familiar to all oceanographers.

To permit exploration of the ocean bottom with continuous rope lengths up to 46,000 feet, required the wire rope be made from wire of approximately 300,000 psi. The wire is stranded and the rope is closed on conventional rope making equipment. But here the differences start. Figure 1-1 is a sketch of this construction. It is difficult to tell by the untrained eye whether the rope is torque balanced or not.

Figure 1-2 explains the principle behind torque balancing. Under tension, each wire in the rope exhibits a torque directly related to the angle it makes with the rope axis. The sum of these torques are equally balanced by the opposing rope torque. When these forces are equal, the rope is virtually nonrotating under a free hanging load. This scientific principle eliminates hockles and kinks caused by unequal torque balance.[1]

To achieve torque balance in rope making, the strand lay is shortened and the rope lay lengthened. The wires are fixed into position by heating the rope to a minimum temperature of 675°F. This process relieves internal stress so the wires stay in the position they are in when heated. Other benefits of stress relieving or thermal stabilization will be discussed.

2.0 TESTING

Oceanographic work is very demanding on a wire rope. It must be as strong as stated, it must not yield prematurely, it must not rotate excessively. Modulus of elasticity is important to length stability.

Certified test reports from the manufacturer should be ordered when the wire rope is procured. The wire rope should be tested using the following procedure.

2.1 Breaking Load - A wire rope requires that a fitting be attached to each end of the sample specimen so it may

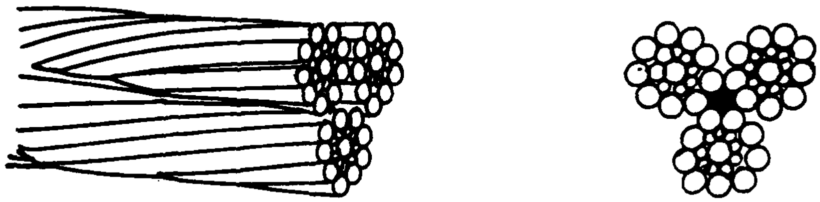


FIGURE 1-1

3 x 19 WIRE ROPE CONSTRUCTION

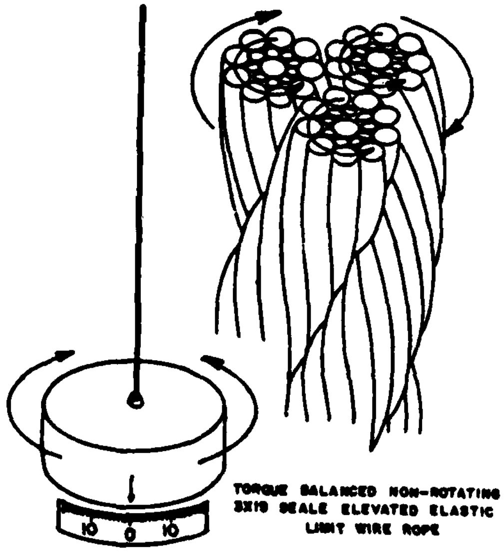


FIG. 1-2
PRINCIPLE OF TORQUE BALANCED WIRE ROPE

be attached to the tensile test machine. Zinc filled wire rope sockets, resin filled wire rope sockets, or swaged sockets are orginarily used. Swaging must be done properly in order to develop full strength. With three-strand wire rope soft annealed wire is inserted in the valleys of the wire rope prior to swaging to improve efficiency. It has been found swaging the full length of the strand rather than in bites, provides a higher breaking strength.

2.2 <u>0.2% Offset Yield</u> - Because of the torque generated in a 6-srand wire rope, it is virtually impossible to determine yield strength by methods used for testing solid bars. Three-strand torque balanced wire rope, however, can readily be tested as if it were a solid bar because of its non-rotating characteristics. The preferred method of test is known as the 0.2% offset yield, the procedure for which is described in ASTM E8. A high yield strength is important in not overloading ropes subjected to repeated shock loads.

2.3 <u>Rotation</u> - The specification for three-strand torque balanced ropes is that it will not rotate more than one degree per foot of rope length with one end fixed and the other end free to rotate. It is important that this specification and the method of test is understood. There have been instances where the rope was tested with both ends fixed or an inoperative swivel placed in the system because the purchase order was not specific.

The correct method of test is .to fix one end in the tensile test machine and attach a ballbearing swivel to the other end before attaching it to the test machine. This end must be free to rotate as the load is applied.

2.4 <u>Modulus of elasticity</u> - Both modulus of elasticity and rotation charateristicS must be known to predict length stability for moorings. Stress relieved or thermally stabilized torque balanced three-strand rope, follows Hooke's Law because of the heat treatment. There is a straight line relationship between stress and strain. Measurement can readily be made from a stress strain curve.

3.0 PROTECTIVE COATINGS

3.1 <u>Galvanize</u> - The wire composing three-strand torque balanced rope has a drawn galvanized coating (amgal). The process wire is first coated with molten zinc in a hot galvanizing process after which it is cold drawn through a series of dies to the required diameter. Although this process reduced the thickness of the zinc, it restores properties to the wire so it meets the same

strength and ductility properties as bright wire. Hot galvanizing provides a metallurgical bond at the zinc-iron interface which is tough and wear resistant. The fatigue life of amgal wire rope in salt water environment is substantially greater than bright rope.

3.2 Plastic Jacketed Wire Rope - Wire rope and strands with extruded plastic coating such as polyethylene provide excellent protection from salt water, salt atmospheres, and chemically corrosive atmospheres. The plastic jacket provides a barrier between the rope and the environment, preventing contact and subsquent corrosion. As added corrosion protection, it is recommended that jacketed ropes be galvanized. Test and field experience have shown that small holidays in the plastic do not destroy the corrosion protection of the jacket. Socket-to-rope interfaces can be fited with boots which will protect this critical area.

Plastic jackets also are beneficial when the major element of corrosion control is being provided by cathodic protection. The cathodic system only needs to protect areas where the plastic jacket is interrupted, thus reducing greatly the area requring protection. The current level and number of anodes required are less with a coating, and the total length protected by one anode is much greater. Tests have shown that plastic-jacketed rope resists axial tensile fatigue better than non-jacketed rope. It is believed the vibration dampening properties of the jacket are responsible for this phenomena. Boots further add to the fatigue life of the product.

3.3 Lubrication - Wire rope is a working machine. Hertzian stresses are extremely high because of point contact between wire and sheave and wire and wire. Lubrication is necessary to minimize wear and the effects of wear. It also acts as a barrier betweeen the steel and sea water to minimize corrosion. Good lubrication is known to double rope life.

4.0 STRESS RELIEVING WIRE ROPE AND IMPROVEMENT OF TECHNICAL PROPERTIES

It is a well known fact that a stress relieved treatment on cold worked steel can remove residual stresses and provide better metallurgical properties of the finished steel product. A problem that needed to be solved was how to stress relieve the long length ropes and strands which needed this treatment. A batch process was judged to be impractical. However, heating of wire rope or strand as it passes through an induction coil has proven to be a satisfactory manufacturing process. In this process, the rope or strand is quickly brought to a stress-

relieving temperature by the induction heating process and then equally as quickly, cooled off by either an air or water quench. This results in very favorable metallurgical properties. It was also noted that wire ropes and strands subjected to this stress relieved treatment, showed preformed charateristics, whether they had been previously mechanically preformed or not. Once this was realized, it became a logical step to stress relieve the finished three-strand torque balanced wire rope. Such a product was found to be fully preformed with no tendency towards twistiness, also no tendency to change the lay and destroy the torque balanced characteristics.

Stress relieving removes residual tensile stresses left in the outer fibers of wires due to the wire drawing process. The combination of these outer fiber tensile stresses, the bending stresses imposed in the wires taking their natural shape in the rope and bending over sheaves or drums, plus the tensile stresses involved in carrying the load result in a very high level of applied stress. This high level of applied stress can lead to micro-cracks and ultimate fatigue failure. The removal of these residual tensile stresses enables the wire to better withstand the imposed loads and thus leads to increased fatigue life. In addition, the wires have increased ductility thus leading to immproved rope breaking strength efficiency. The improvement of wire properties can be seen graphically and explicitly in Figures 1-3 and 1-4. In addition to improved wire properties, improved rope properties were also noted. Increases in elastic limit, yield strength and modulus of elasticity can be clearly seen in Figure 1-5. These increases taken together increase the shock absorbing capacity of the wire rope as measured by the area under the stress curve.

5.0 SPOOLING AND STORAGE OF THREE-STRAND TORQUE BALANCED WIRE ROPE

5.1 Lebus Lagging

The use of Lebus lagging in combination with a diamond cut traverse is preferred when spooling three-strand rope. It can be used with traction winches without developing hockles between the storage drums and haul-off point.

When it is necessary to use a smooth faced drum, the following procedure has been found to be helpful:

5.2 Spooling Three-Strand Wire Rope on Smooth Faced Drum

Multi-layer spooling of thee strand wire rope on smooth-faced drums can be accomplished. This requires uniform distribution of wraps on the bottom layer, without nesting.

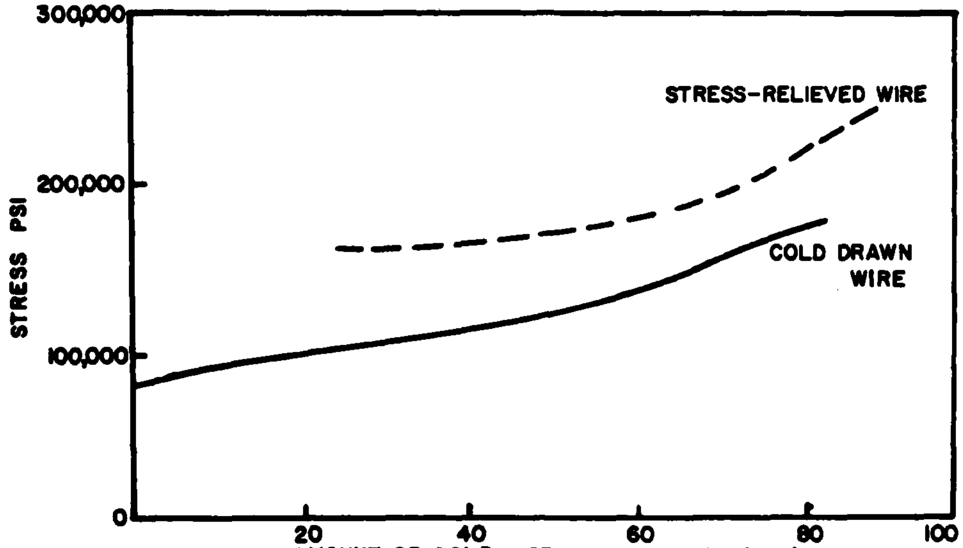


Fig. 1-3 INCREASE IN ELASTIC LIMIT (0.01 % OFFSET)

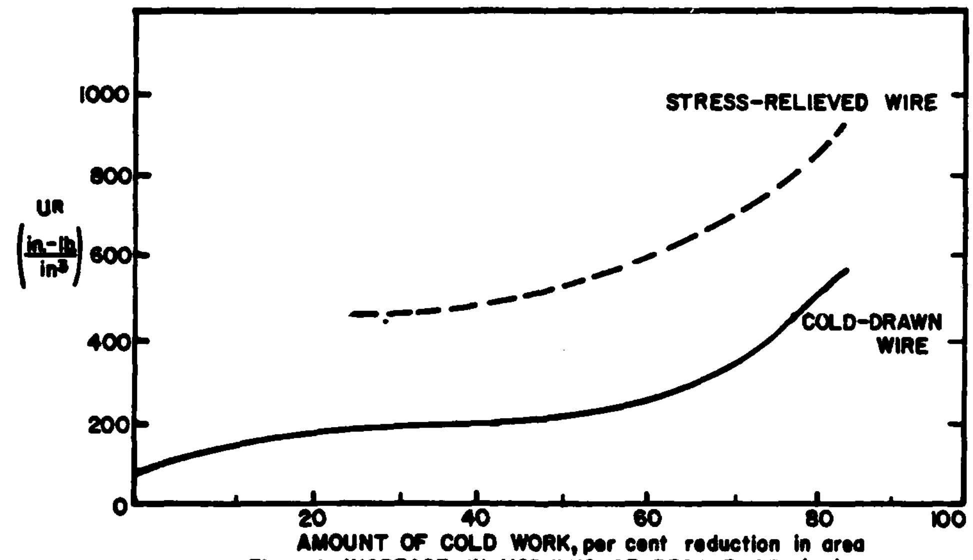
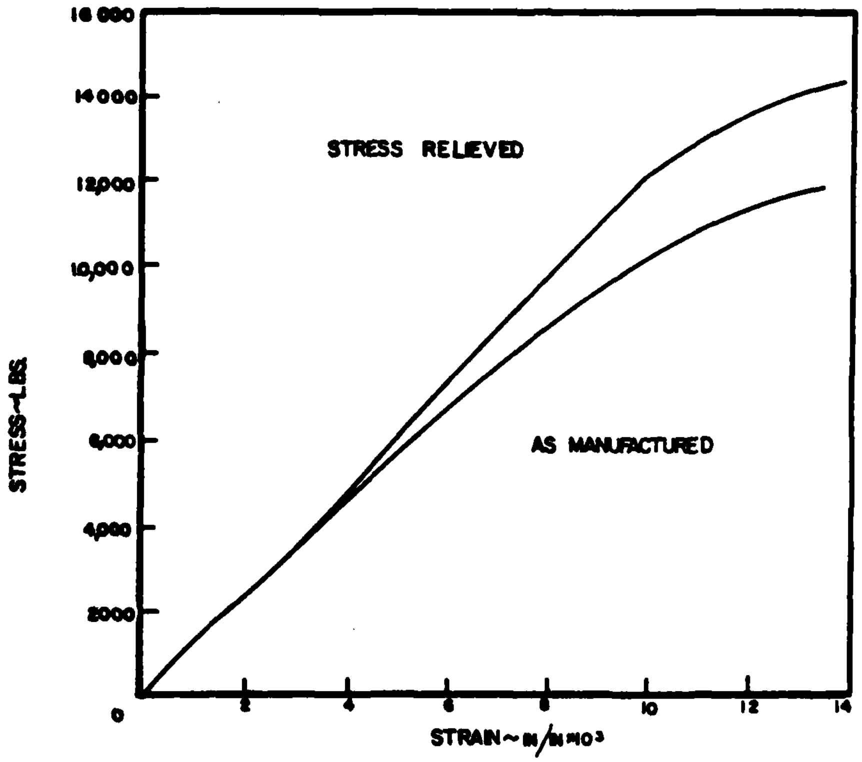


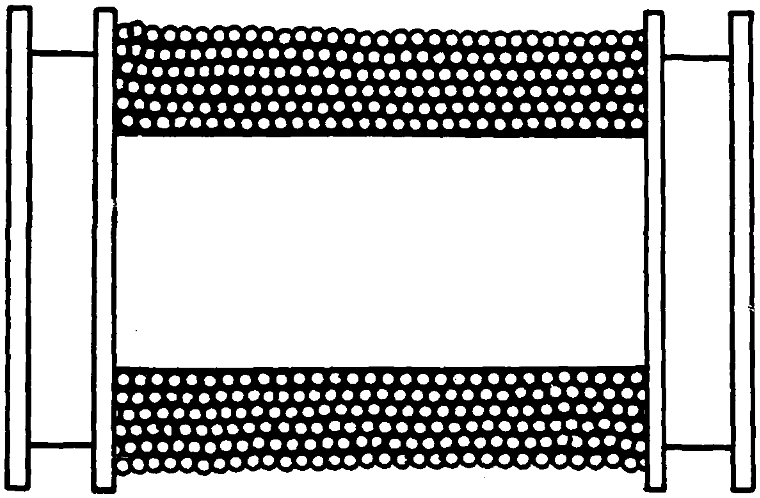
Fig. 1-4 INCREASE IN MODULUS OF RESILIENCE (UR)



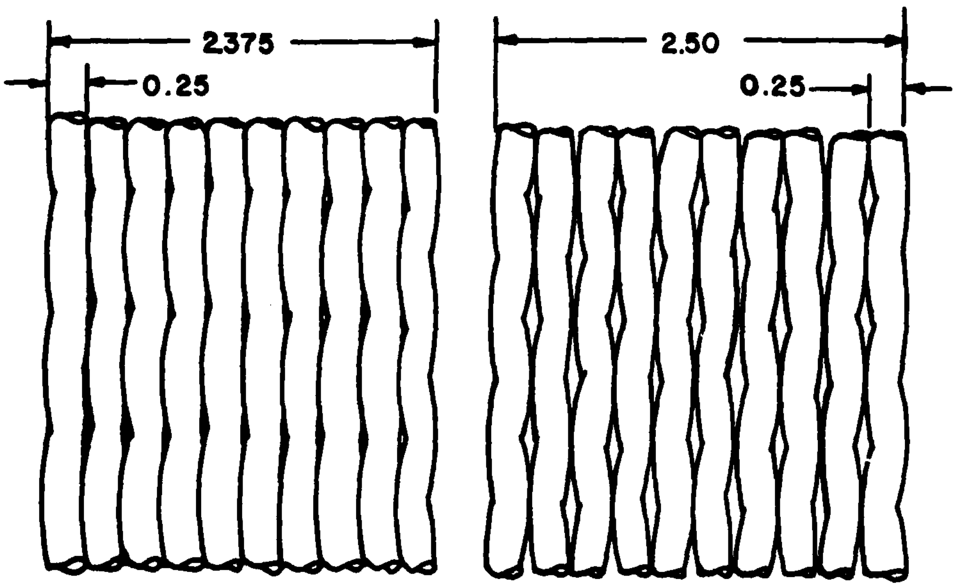
STRAIN IN./IN. x 10^{-3}

	AS MANUF.	STR. RELIEVED
MODULUS of ELASTICITY (PSI)	19,000,000	20,300,000
JOHNSON'S ELASTIC LIMIT (LBS)	9,600	12,400
2% OFFSET Y. S. (LBS)	11,200	14,000
ELONGATION in 24 IN. (%)	3.0	5.1
BREAKING LOAD (LBS)	14,960	15,260

FIGURE 1-5 Improvement in mechanical properties of 3/8" 3 x 19 Seal Torque Balanced Amgal Monitor AA



PROPER WIRE NESTING FIG. 1-6A



10 WRAPS NESTED

10 WRAPS NOT NESTED

EFFECT OF ROPE NESTING ON SMOOTH FACED DRUM. FIG. 1-6B

Nesting permits non-uniform distribution as shown in Figure 1-6. To achieve uniform distribution of wraps with three-strand wire rope, a filler can be inserted between wraps adjacent to the drum.

The size filler to use with three-strand Torque Balanced wire rope is given in Table 1. With this information, the following procedure should be used:

1. Measure width between flanges at drum.
2. Determine how many full wraps can be accommodated with spacing from center to center of wrap being from d_1 to d_2 inches. The closer to d_1, the better the spooling.
3. Insert filler when spooling bottom layer. This is done simultaneously.
4. Practically anything can be used as a filler, but steel strands or IWRC ropes are preferred.
5. The bottom layer must be tight.
6. If a whole number of wraps cannot be accommodated on the bottom layer, a filler or spacer should be added to the flange.
7. After placing the bottom layer, spooling can proceed in normal fashion for parallel grooved drum.

6.0 RECOMMENDED INSTALLATION PROCEDURE FOR TORQUE BALANCED ROPE

These recommendations should be effective for three strand Torque Balanced ropes where the torsional balance is obtained by introducing relatively short strand lays with very long rope lays.

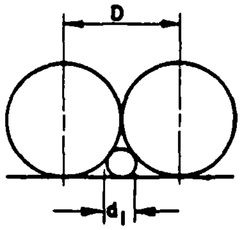
These factors are such that although the rope as manufactured has no twist, or possibly up to one turn of tightening twist, in operating over sheaves the rope develops tendencies to rotate in a direction to unwind more the tightly wound wires in the strands than the long-lay strands in the rope. This results in tendencies of the rope to twist in a tightening direction.

The smaller the sheaves the greater the twist potential; therefore, it is important to provide as favorable sheave-to-rope ratios as is possible.

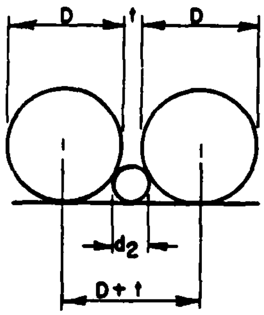
It has been found that in all of our successfully operating Torque Balanced ropes which initially showed tendencies to rotate in tightening direction, the following procedure was followed:

- The dead end of the rope was disengaged from its termination point and approximately five turns of tigthening twist were induced into the rope and the rope reattached to its termination.

Nom Rope Dia., (in.)	Con-struc-tion	Calcu-lated Rope Dia., (in.) D	Spac-ing Toler-ance t	d_1 (in.)	d_2 (in.)
5/32	3 x 7	.173	1/32	.043	.059
11/64	"	.190	"	.0457	.063
3/16	"	.205	"	.051	.067
7/32	"	.233	"	.058	.074
1/4	"	.268	"	.067	.083
5/16	"	.328	"	.082	.097
3/8	"	.395	"	.099	.111
7/16	"	.456	"	.114	.130
1/2	"	.523	"	.131	.146
9/16	"	.589	"	.147	.163
11/64	3 x 19	.190	"	.0475	.063
3/16	Seale	.205	"	.051	.067
7/32	"	.236	"	.059	.075
1/4	"	.265	"	.066	.082
5/16	"	.328	"	.082	.098
3/8	"	.392	"	.098	.114
7/16	"	.460	"	.115	.131
1/2	"	.522	"	.1305	.146
9/16	"	.586	"	.1465	.162
5/8	"	.647	"	.162	.177
3/4	3 x 19	.781	1/32	.195	.211
7/8	Seale	.917	3/64	.229	.253
1	"	1.041	"	.260	.284
1-1/8	"	1.166	"	.2915	.315
7/8	3 x 25 F.W.	.915	"	.229	.252
1	"	1.045	"	.261	.285
7/16	3 x 36 Seale F.W.	.464	1/32	.116	.132
1/2	3 x 46	.528	"	.132	.148
9/16	Seale	.587	"	.147	.162
5/8	F.W.	.645	"	.161	.177
3/4	"	.766	"	.194	.210
7/8	"	.918	3/64	.2295	.253
1	"	1.045	"	.261	.285
1-1/8	"	1.172	"	.293	.316
1-1/4	"	1.314	1/16	.3285	.360
1-3/8	"	1.437	"	.359	.390
1-1/2	"	1.564	"	.391	.422
1-5/8	"	1.718	3/32	.4295	.476
1-3/4	"	1.843	"	.461	.508



$d_1 = \frac{D}{4}$



$d_2 = \frac{D}{4} + \frac{t}{2}$

TABLE II FILLER SIZES FOR SPOOLING 3 STRAND TORQUE BALANCED ROPE ON SMOOTH FACED DRUM

- The hook block was raised and lowered about four times with no pay-loading in order to distribute the induced twist into the rope reeved in the block.
- A pay-load was applied to the hook block, then raised and lowered two or three times. In most situations the hook block operated with no indication of rope twist tendencies. Where twist indications were still evident, an additional three or four twists were induced into the rope at the dead end, with the desired results.

It must be remembered that the smaller the sheaves over which these ropes operate, the greater are the radial pressures of the ropes on the sheaves, the more pronounced is the roller-pin effect on the rope, resulting in greater tendencies of the rope to twist up in a tightening direction.

It should be noted that the direction of twist is in the opposite direction of that encountered in standard hoist ropes operating under the same relative conditions.

Steel reels are much preferred over wood for shipment from the manufacturer, storage, and paying off. The rope should be kept covered and dry. It should not be in contrast with cinders or dirt, as these often contain injurious chemicals. Last but not least, the rope should be lubricated so it is ready for use when used again.

7.0 STAINLESS STEEL TORQUE BALANCED STAINLESS STEEL ROPE (Type 302 and 304)

Wire rope made from types 302 and 304 wire perform well in fresh water. However, in marine environment at ambient temperature, there have been failures which have been the subject of considerable discussion and disagreement. Failure, when caused by corrosion, often is ascribed to stress corrosion, other times to pitting, crevice corrosion or tunnel corrosion.

In some cases, stainless steels are preferred and even necessary when sample cleanliness or hydrogen embrittlement is a factor.

Type 302 and 304 need access to oxygen to maintain a protective oxide film. Moving water over 5 m.p.h. generally will provide such acess.

In Marine environments, salt and marine growth tend to deposit heavily in the valleys between strands thus limiting the oxygen supply in these areas. This leads to a rapid corrosion

of the wire in selected areas and thus the term crevice corrosion. If the depostis in the valleys can be eliminated, rope life should be extended.

Another curious phenomenon is known as tunnel corrosion. This is believed to originate from pits in the steel thus exposing a region having a different electric potential than the steel surface. The pit continues to elongate under the wire surface into hollow tunnels.

Photograph 1 shows an example of tunnel corrosion.

Experiments and experience have shown that stress relieving significantly improved the corrosion resistance of single wires and strands in both marine atmosphere and sea water.

Photograph 2 shows the favorable result of an exposure experiment at the 80 foot International Nickel test lot at Kure Beach, North Carolina.

These results strongly suggest that stainless steel wire ropes have different properties as the result of different processing techniques unique to individual producers. Stress relieving does offer improvement in a marine environment.

Stress relieving also offers an improvement in mechanical properties as shown in the wire rope specifications in a later section.

PHOTOGRAPH 1 (X 15)

Pitting and tunnel corrosion of 1 x 7 strand, constructed from drawn AISI 302 stainless steel wires, caused by exposure in sea water

0.050-inch-diam hard-drawn and stress-relieved AISI 304 wire

0.050-inch-diam hard-drawn AISI 304 wire

0.018-inch-diam AISI 302 hard-drawn wire stress-relieved as stranded

PHOTOGRAPH 2 (X 15)

Appearance of AISI 302 and 304 drawn and subsequently stress-relieved wires and strand after three years of exposure in a severe marine atmosphere

8.0 ENDLESS NASH-TUCK SPLICE FOR THREE-STRAND ROPE

The following dimensions apply to the detailed steps required to make the splice and are denoted in Figures 1 through 5.

Rope Size, Inches	A Feet	B Feet	C Feet	D Inches	E Inches	F Inches
3/16	92	46	45	12	3	5-3/4
7/32	112	56	55	12	3-1/2	6-3/4
1/4	127	61	60	12	4	7-3/4
5/16	163	81-1/2	80	18	5	9-3/4
3/8	193	96-1/2	95	18	6	11-3/4
7/16	224	112	110	24	6-3/4	13-1/2
1/2	254	127	125	24	7-3/4	15-1/2
9/16	254	127	125	24	8-3/4	17-1/2
5/8	254	127	125	24	9-3/4	19-3/8
3/4	256	128	125	36	11-1/2	23-1/4
7/8	256	128	125	36	13-1/2	27-1/4
1	256	128	125	36	15-1/2	31
1-1/8	256	128	125	35	18	34-3/4

- Use "A" length for splicing.
- Unlay both rope ends "B" distance (see Figure 1-7).
- Form marriage at this point and finger lock the three strands of one with the three strands of the other (see Figure 1-8).
- Run back one strand from one end "C" distance, replacing it with the proper strand from the second end (see Figure 1-9).
- Run back another strand from the second end the same distance, replacing it with the proper strand from the first end.
- We now have three pairs of strands protruding from the rope, separated by a distance of "C" (see Figure 1-10).
- Cut off the longer strand ends so that all strands protrude about "D" inches from the rope (see Figure 1-10).
- Unlay each protruding strand "E" distance so that the two strands in each pair protrude "F" apart (see Figure 1-11).

- o Split each protruding strand in half back to point of protrusion so that one-half has a center wire, four inner wires, and the four outer wires covering these inner wires, and the other half has the remaining wires (see Figurs 1-12).

- o When 3 x 7 is used, one-half has the center wire and three adjacent outer wires, and the other half has the four remaining wires.

- o Make ten complete wraps connecting half of one strand with half of the other in each pair (see Figure 1-13).

- o Using a propane torch, anneal all the protruding split strands thoroughly. Be careful and do not heat the live strands.

The desired temperature at which annealing should occur is approximately 1300F (704°C). The use of other than a propane torch using an ambient oxygen suppy should be avoided due to the much greater heat produced by other devices. For example

Torch Type	Tip Temperatue
Ambient O_2 Propane	1700°F (927°C)
Ambient O_2/Propane (Swirl tip)	2700°F (1482°C)
Propane/Oxygen	4579°F (2526°C)
Acc/Oxygen	6000°F (3316°C)

*All temperatures based on medium flame

- o Nash-Tuck each half strand individually with four full tucks by inserting each half strand under the first adjacent strand, over the second, under the third, over the fourth, under the fifth, over the sixth, and under the seventh, following the direction of the crown wires. These half strands must be untwisted at each tuck so that the wires are parallel as they go under and over the rope strands (see Figure 1-14).

- o Pound or press the spliced section into a round tight configuration and cut the protruding split strands off as close to the rope body as possible (see Figure 1-15).

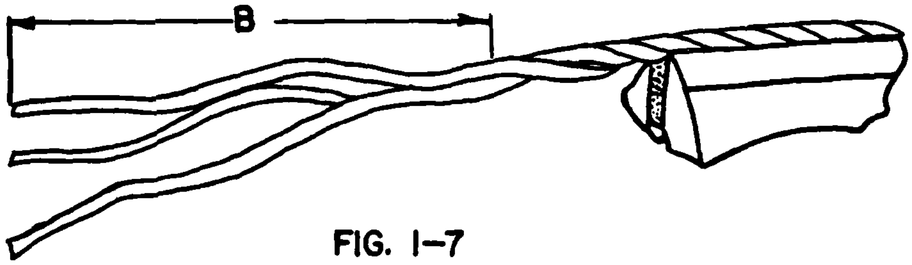


FIG. 1-7

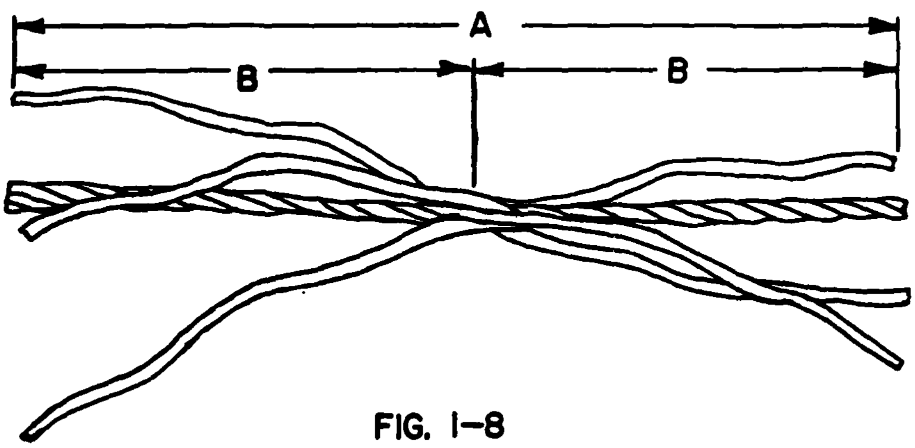


FIG. 1-8

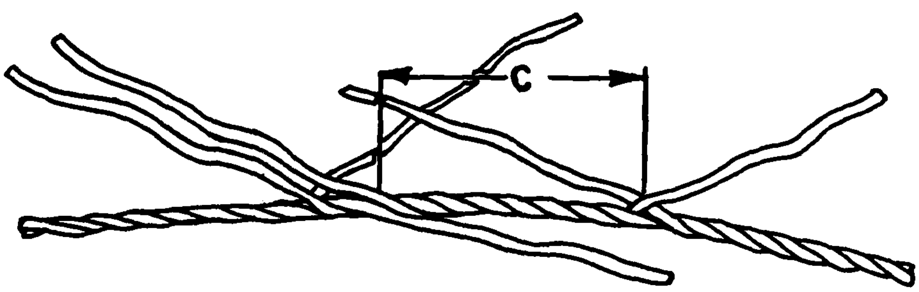


FIG. 1-9

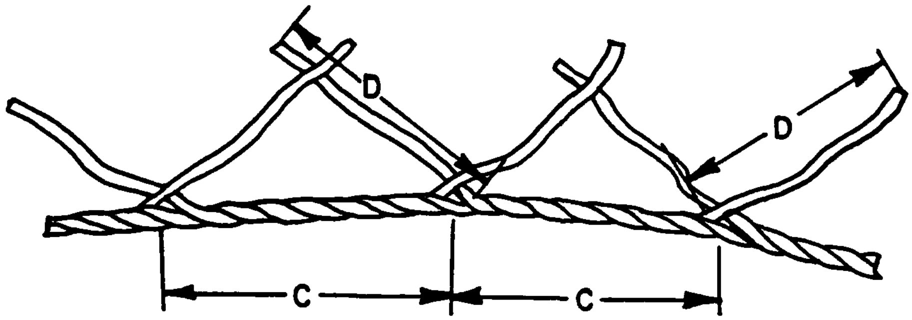


FIG. 1—10

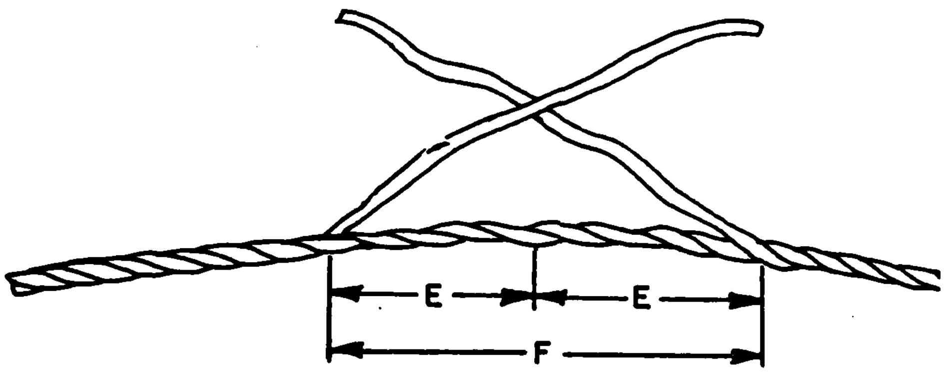


FIG. 1—11

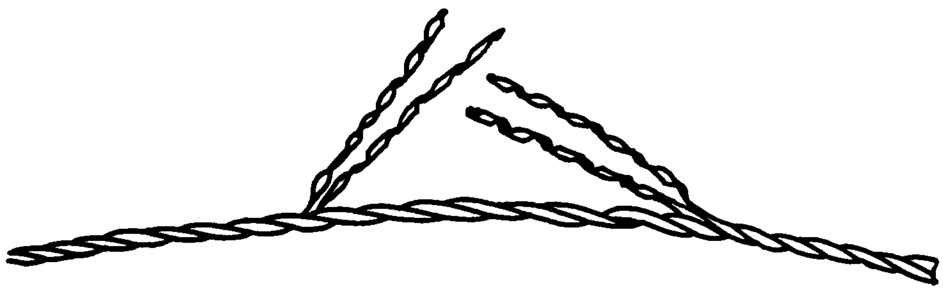


FIG. 1—12

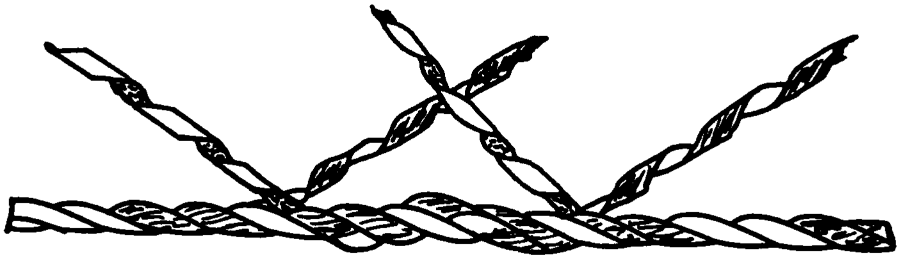


FIG. 1—13

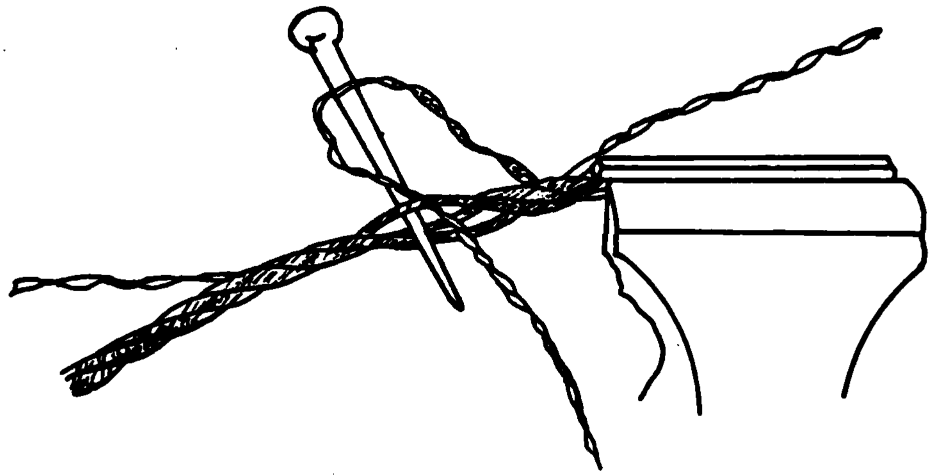


FIG. 1—14

FIG. 1—15

Caution, spliced 3 x 19 wire rope should be rated at 80% of its original, new rope breaking strength. This percentage is based on a properly installed Nash_Tuck splice as described above. Shorter splices than recommended, improper technique or accidental annealing of the rope itself will further reduce the efficiency of the splice.

9.0 WIRE ROPE RETIREMENT CRITERIA

A wire rope should be considered for replacement if it has three or more broken wires per strand in one strand lay or six or more broken wires in one rope lay.

It is important the wire surface be sufficiently clean so that the broken wires are visible. A good inspection method is to encircle rope with a rag or cotton waste and run the hoist slowly. If broken ends protrude and catch the waste or ends, bits of it will show location of the broken wires. It is best to face in the direction the rope is moving when holding the rag so it is pulled away from the holder if it snags on broken wires. A bare or gloved hand, rather than a rag or cotton waste, can be dangerous. A rope speed of 50 fpm or less is usually suggested.

Diameter and lay length measurements are most easily made at the same time and at the same location along the rope. A significant reduction in diameter can be the result of loss of metallic area from corrosion or from stretching due to broken wires.

Strand rope should be measured with a three-point micrometer to obtain a meaningful measurement. Look for corrosion at and under attachments and at the end terminations. Corrosion is a reason for replacement as it is difficult to estimate remaining strength.

Structural damage is fairly easy to see. Types that call for immediate rope removal, if they cannot be removed by cutting off the ends of the rope include kinks and doglegs.

Some additional, but more subtle forms of wire damage which will effect wire performance and ultimate strength are abrasions of the outer wires and general corrosion; i.e., flaking rust. The flattening of external wires due to abrasion occuring at the outboard sheave, level wind rollers or by the wire coming into contact with a fixed object reduces the wire load carryng capacity and should be watched for during visual inspections of the wire.

A wire condition exhibiting flaking rust indicates that the wires protective zinc coating has been lost and that the individual wire in the rope is being reduced through an oxidization process. This results in a reduction of the metalic area

of each strand with a resulting lowering of the ultimate strength of the whole wire. Since this condition is difficult to evaluate for the entire rope length a rope exhibiting this condition over a major portion of its length should be retired from service.

10.0 MINIMUM TREAD DIAMETERS OF SHEAVES AND DRUMS IN INCHES FOR THREE-STRAND WIRE ROPE

Rope Dia.	3 x 7	3 x 19	3 x 25	3 x 46
1/4	14.5	11.5		
5/16	19	14.4		
3/8	23	17		
7/16	27	20		
1/2	29	23		13
9/16	35	26		15
5/8	38	29		16
3/4	46	35	28	
7/8	54	40	33	23
1	58	46	37	26
1-1/8		52	42	30
1-1/4		58	47	33
1-3/8		64	52	36
1-1/2		70	57	40
1-5/8				44
1-3/4				47
1-7/8				50
2				53

These minimum tread diameters are based on factors of approximately 400 times the diameter of outer wires.

11.0 GROOVES

Grooves in sheaves and drums should be slightly larger than the rope, in order to avoid pinching and binding of the strands, and to permit the rope to adjust itself to the radius of curvature. The greater the angle of approach to the groove, the larger the tolerance required to prevent excessive flange wear (Figure 1-16).

The diameter of an unused rope may exceed the nominal diameter by the amounts specified in the following table:

Diameter Tolerances for Wire Rope

Nominal Diameter of Rope in inches	Undersize %	Oversize %
0 to 1/8	0	8
Over 1/8 to 3/16	0	7
Over 3/16 to 1/4	0	6
Over 1/4	0	5

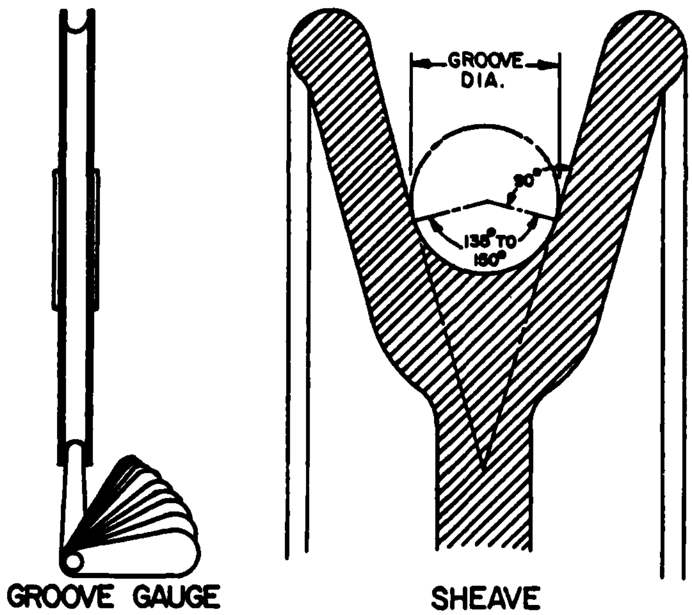


FIGURE 1-16 SHEAVE GROOVE DIMENSIONS

Grooves which have been worn to the minimum diameter shown in the table should be remachined to the minimum diameter shown for New or Remachined Grooves. Grooves of too large diameter do not properly support the rope, and permit it to become elliptical.

Tolerance Groove Diameter Should Exceed Nominal Rope Diameter

Nominal Diameter of Rope in Inches	Minimum (%)	New or Remachined Grooves (%)
0 to 1/8	4	8
Over 1/8 to 3/16	3.5	7
Over 3/16 to 1/4	3	6
Over 1/4	2.5	5

12.0 FLEET ANGLE

On installations where the wire rope passes over a lead sheave then onto a drum, it is important that the lead sheave be located at a sufficient distance from the drum to maintain a small fleet angle at all times. The fleet angle is the side angle at which the rope approaches the sheave from the drum. It is the angle between the center line of the wire rope.

Experience has proven that the best wire rope services is obtained when the maximum fleet angle is not more than 1-1/2 degrees for smooth drums and 2 degrees for grooved drums. The maximum fleet angle is an angle between the center line of the sheave and the rope when it is at the end of its traverse travel on drum. Fleet angles of 1-1/2 and 2 degrees are the equivalents of approximately 38 and 29 feet, respectively, of lead for each foot of rope traverse travel either side of the center of

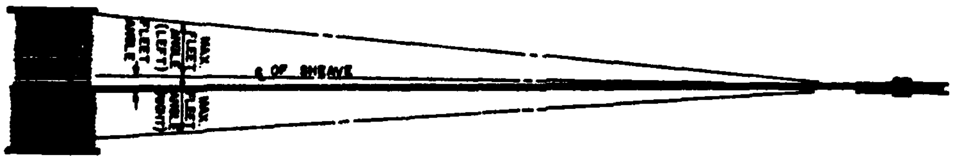


FIG. 1-16 EXAMPLE OF FLEET ANGLES

travel in line with the lead sheave should be located not less than 57 feet from the lead sheave. If the drum were grooved, the minimum distance should be approximately 43.5 feet.

13.0 HOW TO ORDER - WIRE ROPE SPECIFICATIONS

To ensure that your order for three-strand wire rope is accurately filled, the following information should be included for each item.

- o Length: The length of each piece and number of pieces should be specified.
- o Diameter: Diameter should be specified.
- o Construction should be specified.
- o Finish: Amgal should be specified for carbon steel. Stainless steel would ordinarily be furnished bright.
- o Grade: The grade should be stated, such as Monitor AA (Extra Improved Plow), Monitor AAA, etc.
- o Preforming: When excellay preforming is desired, the order should so state.
- o Lay: Torque Balanced should be specified. Right lay will be furnished unless specified otherwise.
- o Rotation Resistance: Maximum rotation of 1 degree per foot of length when loaded to 70% of strength with one end fixed and the other free to rotate.
- o Special Processing: Stress relieving or elevated elastic limit should be specified.
- o Plastic Jacketing: Specified when required.
- o Fittings: Details as to type attachment would be specified when required.
- o End Termination.
- o Modulus of Elasticity: Should be specified when length stability is important. 3 x 19 torque balanced elevated elastic limit 20,000,000 psi minimum.

14.0 NON-DESTRUCTIVE TESTING

Electromagnetic non-destructive testing of wire rope has been in use for over 25 years. During this period, more and more extensive use of this method has been relied on to evaluate the material condition of wire ropes used in situations where personnel or equipment safety are concerned, i.e., mining and the ski industry. Normally such inspections are carried out by agencies whose trained personnel are expert in the interpretation of signals resulting from broken or corroded wires.

Basically, the testing units are either AC or DC in nature. The AC unit measures the differences in cross sectional areas and is best for corrosion detection. The DC units are best used to detect broken wires if sufficient separation of the broken ends.exists. However, cracked wires cannot normally be detected with this unit. The principals of operation for both individual AC/DC test units and the unitized AC/DC units is discussed in the following text.

14.1 Individual AC/DC Units

DC Unit: In the DC unit, strong permanent magnets are placed around a section of wire rope so that the rope becomes saturated with magnetic lines of flux (Figure 1-18). Lines of flux can be observed by iron filings sprinkled on top of a piece of paper having magnets underneath (Figure 1-19). The flux appears to "flow" from the north to south pole; however, the lines are stationary. The lines of flux are also distinct because of the existence of both attractive and repulsive forces. Saturation means that if stronger magnets were used, the number of lines of flux for a given cross section (flux density) would remain essentially unchanged.

If a broken wire were present in a saturated section of rope, then a north and south pole would be formed and the lines of flux would "jump" the gap (Figure 1-20). It is these lines of flux, called flux leakage, that can be detected to indicate a broken wire. Pitting from corrosion and localized wear will also interrupt the saturated lines of flux and cause flux leakage.

A classic physics experiment is to demonstrate that a magnetic field can produce an induced voltage in a conductor that is passed through the magnetic field. The conductor, passing at right angles through the lines of flux, must have a minimum travel speed through the flux field in order for the voltage to be large enough to measure (Figure 1-21).

Flux leakage in the wire rope is detected by using this phenomenon. However, in this case, the conductor is a search coil that is held stationary while the magnetic field is moving. In the inspection equipment, search coils are placed around the

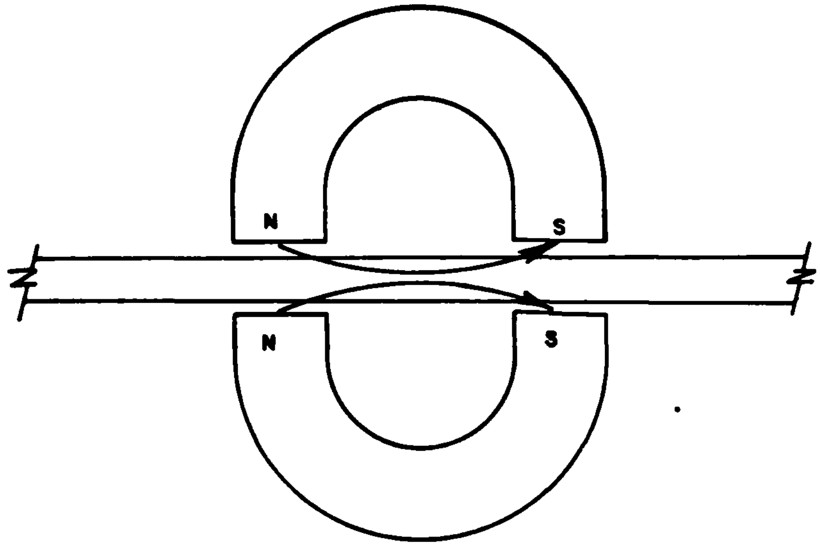


FIG. 1—18
SATURATION OF WIRE ROPE WITH LINES OF FLUX.

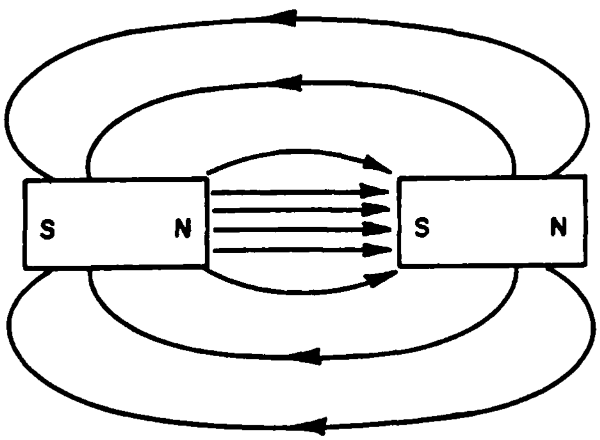


FIG. 1—19 LINES OF FLUX

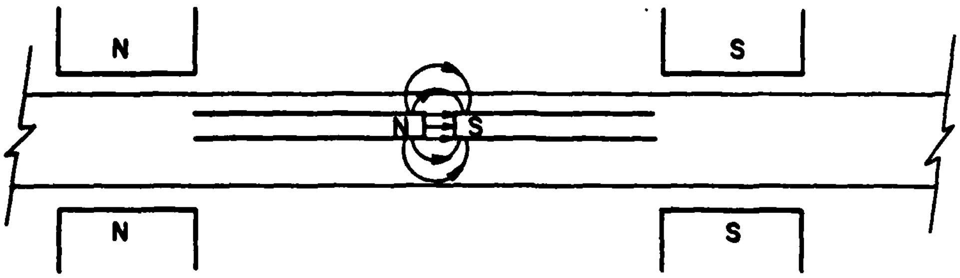


FIG. 1-20 FLUX LEAKAGE LOCATION OF BROKEN WIRE

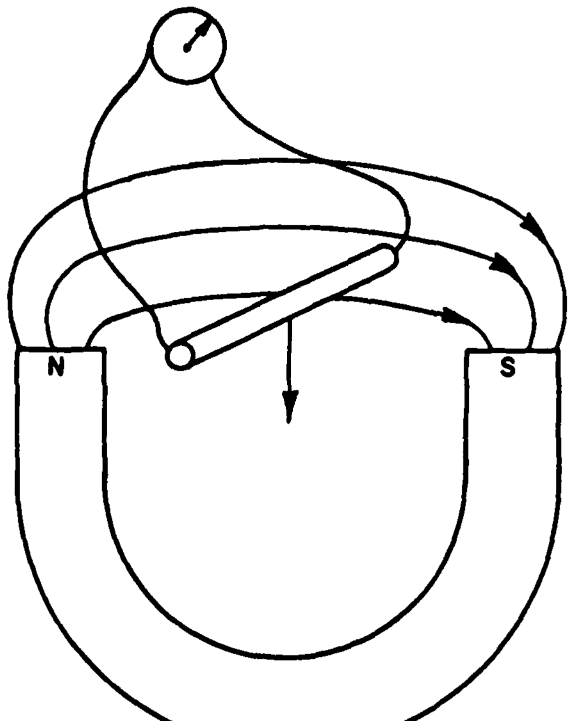


FIG. 1-21 A CURRENT IS CREATED IN A CONDUCTOR MOVING THROUGH LINES OF FLUX

saturated wire rope between the poles of the permanent magnet. The rope travels at some minimum speed; thus, any flux leakage will also be moving and will pass through the search coils (Figure 1-22). When this occurs, an induced voltage is generated in the search coils, and, by proper amplification and conditioning of the signals, the broken wire is detected.

For the DC unit, there must be relative motion between the sensor coil and the wire rope. This means that the rope must travel through the sensor head or, for a stationary rope, the sensor head must travel along the wire rope. A minimum velocity of about 50 fmp is required.

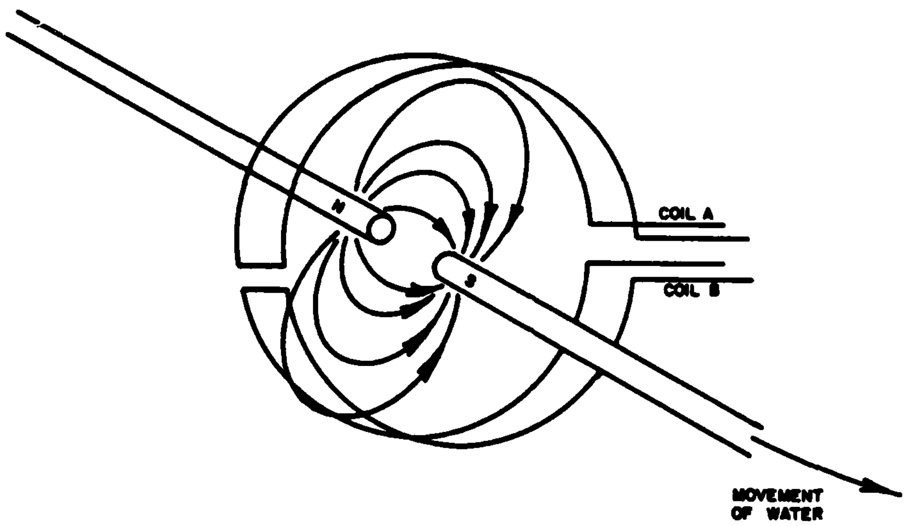


FIG. 1-22 SEARCH COILS PICK UP RADIAL COMPONENT OF FLUX LEAKAGE

Below this speed the induced voltage in the sensor coil is too small to detect broken wires. The velocity must also remain constant for signal strength to be consistent; however, to account for changes in velocity, the DC unit is built with a tachometer coupled to an amplifier so that signal strength can be amplified for changes in velocity.

Two search coils are usually built into the sensor head, as shown in Figure 1-22, to allow the head to clamp around the rope. Data output can take several forms because signals from two search coils are available. Usually two output traces are shown so signals from coil A and B can be displayed as a combination of A, B, A + B, $(A + B)^2$ or AB. Typically the data is displayed as A + B on one trace and $(A + B)^2$ on the second trace.

AC Unit: A relatively weak alternating magnetic field is produced by electromagnets in an AC unit sensor head. These magnets function as the primary coil of a transformer (Figure 1-23). The wire rope serves the purpose of the ferromagnetic core of a transformer. A secondary coil in the middle of the sensor head produces an output voltage that is proportional to the magnetic flux "flowing" through the wire rope. Variations in the cross-sectional area of the wire rope influences the strength of the magnetic flux field and, thus, the strength of the output voltage. Hence, loss of metallic area can be measured by the output voltage.

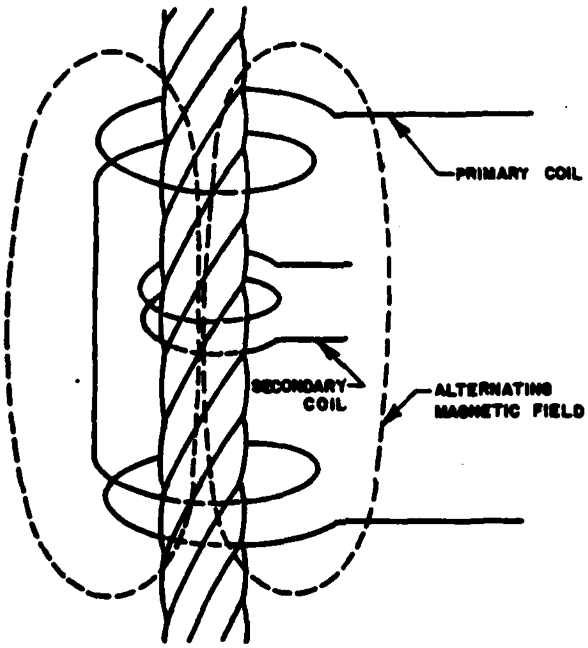


FIG. 1-23 THE BASIC PRINCIPLE OF THE AC METHOD

The sensor coil measures the metallic volume over a 2- to 3-inch length of wire rope. Wear and corrosion can produce a significant volume change within the finite length, but a single broken wire with a small gap between the ends reduces the volume insignificantly. If many breaks occur within the finite length or a wire is missing, then a defect signal may be recorded.

In the AC unit, the magnetic flux field always varies with time because of the alternating field. Hence, a voltage is produced in the sensor coil whether or not the rope moves.

Because of the alternating magnetic field, small electric currents are induced that circulate around the rope axis

within and between the wires. These eddy currents also alternate and produce their own magnetic fields which tend to oppose that from the primary field. This opposition produces a phase shift between the peaking of the magnetizing current and that of the sensor coil voltage. Built-in circuits in the instrumentation utilize the phase shift to produce a second data trace. The first data trace, called X, is essentially proportional to the axial component of the flux field in the rope, and therefore, measures loss of metallic area. The second trace, called R, is proportional to the magnitude of the eddy currents and reflects conditions within the rope that cause changes in the eddy currents. Corrosion products or lay tightening or loosening will affect the passage of eddy currents. Thus, by comparing the X and R traces, wear and corrosion can usually be differentiated.

14.2 Unitized AC/DC Unit

The unitized AC/DC unit uses a sensor head having strong permanent magnets to saturate the wire rope with magnetic flux. This is similar to the individual DC unit; however, the means of sensing the faults in the rope is different. Hall effect sensors are used to detect faults.

Hall effect sensors are solid state devices which can detect and accurately measure magnetic fields. Figure 1-24 shows a sketch of a sensor. Electrical wires are bonded to all four sides of a semiconductor chip. A constant current is passed between two opposing edges. The other two edges develop a potential difference when the semiconductor chip is placed in a magnetic field. The potential difference developed by the sensor is directly related to the strength of the flux field. Static magnetic fields can be measured; this is a feature not available in the individual AC/DC units. This means that ropes traveling at extremely slow speeds can be inspected, which is desireable when trying to pinpoint a fault location.

Figure 1-25 shows the placement of Hall effect sensors in the Magnograph sensor head which clamps around a wire rope. The Hall effect sensors located between the poles of the magnets pick up flux leakage, which indicates broken wires or other local faults (or LF). The Hall effect sensors at the poles of the magnets measure the quantity of flux "flowing" into the wire rope. When the cross-sectional area of steel changes, so does the flux "flowing" into the rope; thus, loss of metallic area (or LMA) is measured.

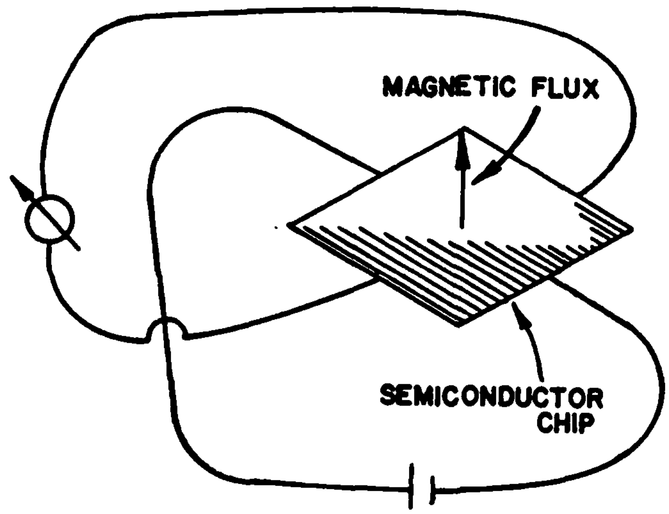


FIG. 1-24 SKETCH OF A HALL EFFECT SENSOR.

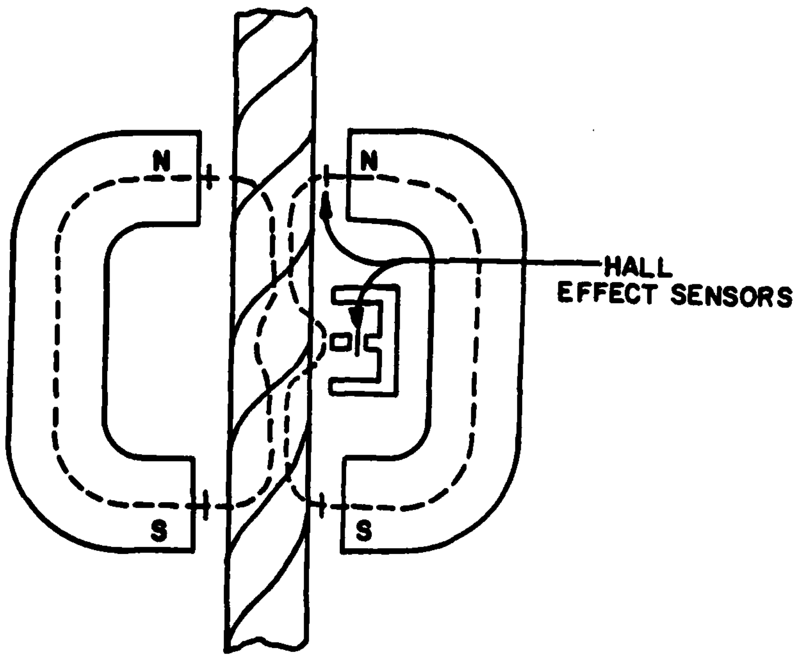


FIG. 1-25 HALL EFFECT SENSOR PLACEMENT FOR THE MAGNOGRAPH.

REFERENCES

Robeling Wire Rope Handbook. Copyright 1966, The Colorado Fuel and Iron Corporation.

H. H. Haynes and L. D. Underbaake. Technical Note, TNN-1594, Civil Engineering Laboratory, Naval Construction Batallion Center, Port Hueneme, October 1980.

Wire Rope Engineering Handbook, U. S. Steel Supply.

CHAPTER 2

Oceanographic Electro-Mechanical Cables

ALBERT G. BERIAN

1.0 CONSTRUCTION CHARACTERISTICS	2-4
1.1 COINCIDENCE	2-4
1.2 CENTER STRENGTH MEMBER	2-4
1.3 BRAIDED OUTER STRENGTH MEMBER	2-5
1.4 ELECTRO-MECHANICAL WIRE ROPE	2-5
1.5 OUTER SINGLE SERVED STRENGTH MEMBER	2-6
1.6 OUTER DOUBLE SERVED STRENGTH MEMBER	2-6
1.7 3-4-5 LAYER SERVED STRENGTH MEMBER	2-7
2.0 WORKING ENVIRONMENT	2-7
2.1 FLEXING	2-7
2.2 ABRASION	2-7
2.3 TENSION CYCLING	2-8
2.4 CORROSION	2-8
2.5 FISH BITE	2-9
2.6 ABRASION RATE FACTORS	2-9
2.7 KINKING	2-9
2.8 CRUSHING	2-9
3.0 PARTS OF CONTRA-HELICALLY ARMORED E-M CABLE	2-9
3.1 DIRECTION OF LAY	2-10
3.2 LAY ANGLE	2-10
3.3 PREFORM	2-11
3.4 HEIGHT OF HELIX	2-11
3.5 PERCENT PREFORM	2-12
3.6 LENGTH OF LAY	2-12
3.7 PITCH DIAMETER	2-12
3.8 NUMBER OF ARMOR WIRES	2-13

3.9 ARMOR COVERAGE 2–14
3.10 SQUEEZE 2–14
3.11 CORE 2–15
3.12 WATER BLOCKED CORE 2–15
4.0 PERFORMANCE CHARACTERISTICS OF C-H-A,E-M CABLES 2–16
4.1 TORQUE BALANCE 2–16
4.2 TWIST BALANCE 2–17
4.3 CRUSH RESISTANCE 2–17
4.4 CORROSION RESISTANCE 2–18
4.5 ABRASION RESISTANCE 2–19
4.6 ELONGATION 2–20
4.7 SEA WATER BUOYANCY 2–20
4.8 BREAKING STRENGTH 2–20
5.0 MANUFACTURING PROCESSES FOR E-M CABLES 2–22
5.1 CONDUCTOR STRANDING 2–22
5.2 INSULATION 2–22
5.3 WET TEST 2–22
5.4 CABLING 2–23
5.5 BRAIDING 2–23
5.6 SERVING 2–23
5.7 JACKETING 2–23
5.8 ARMORING 2–24
5.9 PRESTRESSING 2–24
6.0 HANDLING E–M CABLES 2–27
6.1 STORAGE BEFORE USE 2–27
6.2 SPOOLING EFFECT ON E–M CABLES 2–28
6.3 SMOOTH DRUM SPOOLING 2–29
6.4 TENSION SPOOLING OBJECTIVES 2–29
6.5 TENSIONS FOR SPOOLING 2–29
6.6 LOWER SPOOLING TENSIONS 2–30
6.7 GROOVED DRUM SLEEVES 2–30
6.8 SHEAVES 2–30
7.0 FIELD INSPECTION AND TESTING 2–31
7.1 GENERAL 2–31
7.2 REQUIRED INSPECTIONS 2–34
7.3 CABLE RECORD BOOK 2–34

	7.4 CABLE LOG	2-34
	7.5 INSPECTION	2-34
	7.6 VISUAL INSPECTION PRACTICES	2-35
	7.7 ARMOR TIGHTNESS INSPECTION	2-35
	7.8 LAY LENGTH OF THE OUTER ARMOR	2-37
	7.9 CONDUCTOR ELECTRICAL RESISTANCE	2-39
	7.10 OUTSIDE DIAMETER	2-39
	7.11 NEED FOR LUBRICATION	2-40
	7.12 LOCATION OF OPEN CONDUCTOR	2-43
	7.13 FAULT LOCATION	2-44
	7.14 RE-REELING	2-44
	7.15 CABLE LENGTH DETERMINATION	2-45
8.0	RETIREMENT CRITERIA	2-45
	8.1 CONSIDERATIONS	2-45
	8.2 BROKEN WIRE CRITERIA	2-46
	8.3 LIFE CYCLE CRITERIA	2-47
	8.4 NON-DESTRUCTIVE TESTING	2-48
9.0	CABLE MATERIALS	2-48
	9.1 CONDUCTORS	2-48
	9.2 INSULATIONS	2-50
	9.3 SHIELDING	2-50
	9.4 JACKETS	2-51
	9.5 ARMOR	2-51
10.0	CONTRA-HELICALLY ARMORED E-M CABLE SPECS	2-53
	10.1 PERFORMANCE vs. CONSTRUCTION SPECIFICATION	2-53
	10.2 CONSTRUCTION SPECIFICATION	2-53
	10.3 PERFORMANCE SPECIFICATION	2-54
11.0	AVAILABLE CABLE SERVICES	2-55
	11.1 GENERAL	2-55
	11.2 SPOOLING	2-57
	11.3 SPLICING	2-57
	11.4 FAULT LOCATION	2-57
	11.5 RECONDITIONING	2-57
	11.6 MAGNETIC MARKING	2-58
12.0	APPENDICIES	2-61
13.0	BIBLIOGRAPHY	2-94

1.0 CONSTRUCTION CHARACTERISTICS

Eletro-mechanical (E-M) cables constitute a class of tension members which incorporate insulated electrical conductors. The spacial relationship of these two functional components may be:

1.1 Coincident (Figure 2-1), as in an insulated, copper-clad steel conductor conventionally used in sonobuoy and trailing cables of wire-guided missiles.

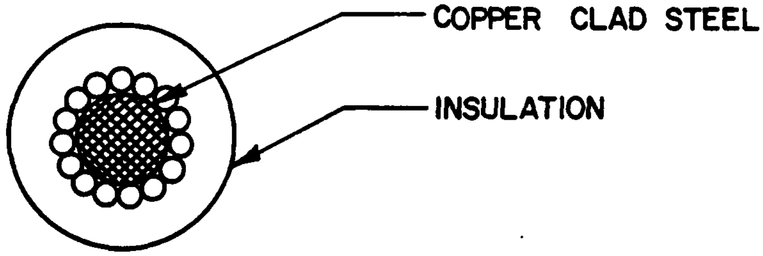


FIG.2-1 COINCIDENT STRENGTH-MEMBER AND ELECTRICAL CONNECTION

1.2 Center Strength Member (Figure 2-2), such as for elevator traveling control cables. In this, as in most constructions wherein the strength member and electrical component are separate elements, the strength member may be one of several metals or non-metallic materials. Also, the construction of the strength member may be a solid but more generally, it is a structure of metal or yarn filaments. The electrical components of the cable are arranged around the strength member and an outer covering jacket is usually used.

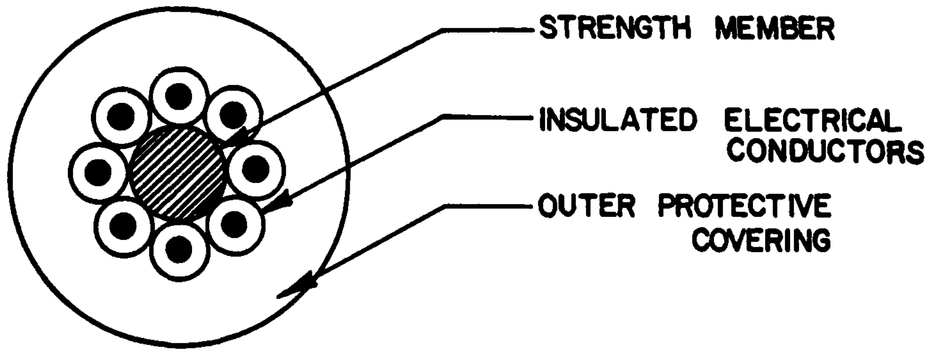


FIG.2-2 CENTER STRENGTH-MEMBER E-M CABLE

1.3 Braided Outer Strength Members (Figure 2-3), involve a center arrangement of electrical conductors (one, coax, tristed, pair, triad, etc.) with the braided metal or non-metal strength member external to the elecrical conductors. Because of the mechanical frailty of the relatively fine filaments a protective covering or jacket is usually required.

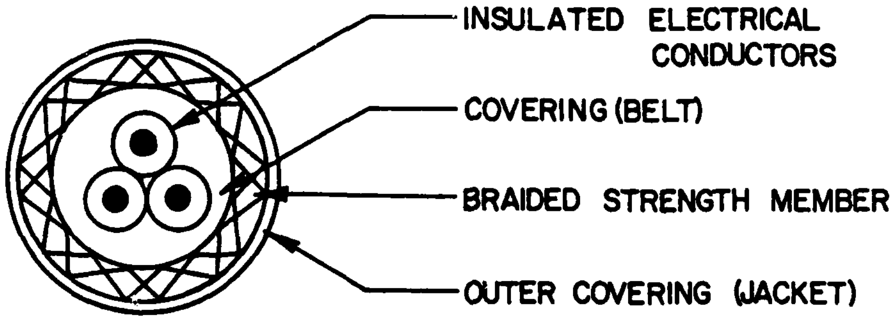


FIG.2-3 BRAIDED OUTER STRENGTH MEMBER

1.4 Electro-Mechanical Wire Rope (Figure 2-4), uses standard wire rope constructions; a three-strand is illustrated. The insulated electrical conductors can be located in two parts of the cross section, in the strand core and in the outer valleys or interstices. When conductors are placed in the outer interstices, a protective covering, or jacket is needed.

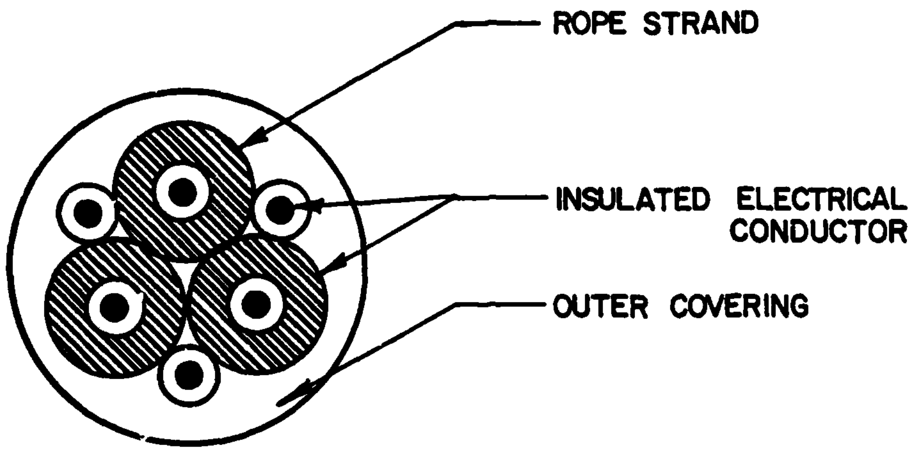


FIG.2-4 ELECTRO-MECHANICAL WIRE ROPE

1.5 Outer Single Served Strength Member (Figure 2-5), utilizes metal or non-metal fibers which are helically wrapped around the electrical core which contains the insulated electrical conductors. The metal or non-metal fibers are helically wrapped around the electrical core so that they completely cover the surface. Because this construction has a high rotation vs tension charateristic, it is impractical as a tension member; the wrapping being used to increase resistance to mechanical damage.

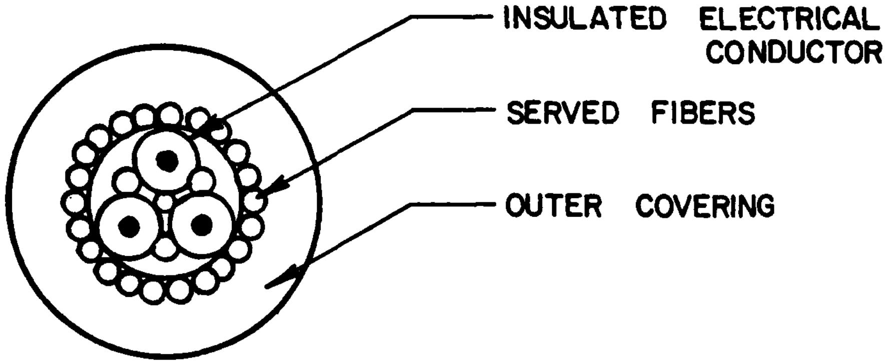


FIG.2-5 SINGLE SERVED STRENGTH-MEMBER E-M CABLE

1.6 Outer Double Served Strength Member (Figure 2-6), has two helical serves of metal or non-metal fibers which are wrapped around the electrical core. The two helical wraps are usually served in opposite directions to obtain a low torque or low rotation vs tension performance characteristic. An outer covering may be used; its purpose being primarily corrosion protection.

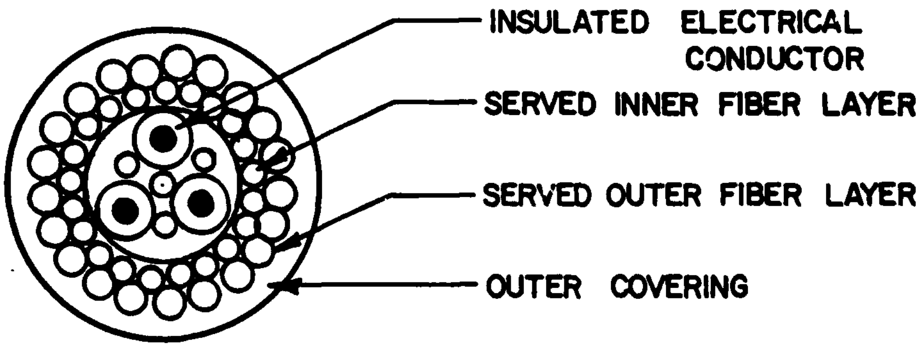


FIG.2-6 DOUBLE SERVED STRENGTH-MEMBER E-M CABLE

1.7 3, 4, 5 Layer Served Strength Member (Figure 2-7), utilize more layers of the served strength member to increase the ultimate tensile strength, or breaking strength of the E-M cable. The direction of helical serve for a three-layer serve is, from inner to outer serve, right-right-left. For a four-layer serve the directions are left-right-right-left.

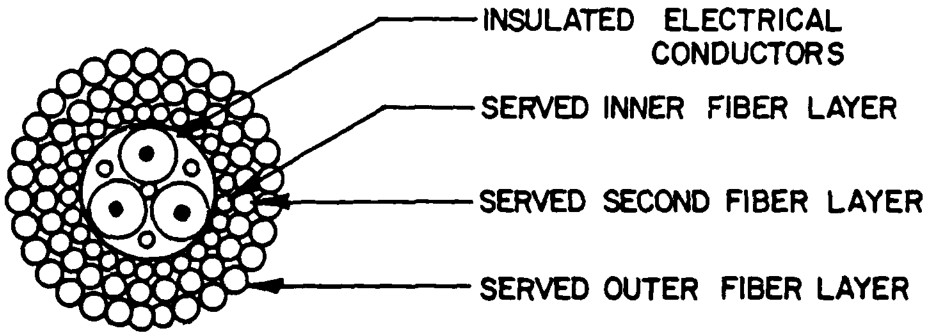


FIG. 2-7 TRIPLE SERVED STRENGTH-MEMBER E-M CABLE

2.0 WORKING ENVIRONMENT

In the above discussion of constrution of E-M cables, no mention was made of the working environment, which for this discussion is oceanographic.

The hazards of this environment, which are important to E-M cables, include:

2.1 Flexing - In most applications operating from ships there is constant motion in service with resulting bending of the E-M cable at points of changing direction such as on sheaves, fairleads, winch drums, capstans, level winds, motion compensators, etc.

2.2 Abrasion - This motion results in the development of two forms of abrasion; between cable internal components and external between the cable and the handling equipment. This abrasion degradation can progress to a point where either a failure occurs or it is observed to be unfit for continued use and is retired from service. The latter is, of course, the more desirable approach.

The rate of abrasive wear varies with serveral operational factors including line speed, tension, and bend diameter as a ratio of cable diameter. Also, maintenance factors such as allowing abrasive materials (sand, corrosion, etc.) to remain

in the cable and maintaining the proper lubrication of rubbing metal parts have a significant affect on the deleterious effects of flexing.

2.3 Tension Cycling - When deployed from a moving platform, the tension in the E-M cable will vary constantly. The magnitude of the tension variations can be reduced by use of such devices as motion compensators. Because the E-M cable is an elastic member, it has a tension/elongation characteristic defined by its elastic modulus (see Appendix 10). As the magnitude of stretch varies, the compoents change geometrical relationship and create internal friction very much similar to that in flexing. The same damage alleviating and enhancing factors apply as for flexing conditions.

2.4 Corrosion - Applying to metal, primarily steel, this is a major concern in the marine environment. Galvanized steel is, because of its low life cycle cost, the most common metal used for the very common double layer armored cables. The galvanized coating, usually about 0.5 ox./ft^2, is usually electrolytically dissolved very quickly leaving basic steel to be attacked by the sea water. Figure 2-8 shows the equivalent thickness to be about .000500 inch. Using an average surface reduction by corrosion for steel of 0.001 inch per year, this thickness would be completely eliminated in six months.

Figure 2-8

Thickness of Zn Coating on GIPS Armor Wires

$$\text{Usual specification} = 0.5\ \frac{02}{ft^2} =$$

$$= \frac{0.502}{(16\frac{02}{lb})\ ft^2} = .03125\ \frac{lb}{ft^2}$$

$$\frac{t}{23} = .02135\ \frac{lb}{ft^2} \qquad t = \text{thickness (in.)}$$

$$= \text{density}\ \left(\frac{lb}{ft^3}\right)$$

$$t = \underline{12 \times .03125}$$

$$\text{for Zn} = 12 \times 62.4\ \frac{lb}{ft^3}$$

$$t = \frac{12 \times .03125}{12 \times 62.4} = .000500\ \text{inch}$$

2.5 Fishbite - This hazard applies to cables having an outer surface which is soft relative to steel. This class of cables include those with extruded outer coverings, or jackets, and those having a covering of braided yarns such as polyester and aramid.

2.6 Abrasion Rate Factors - The rate of this degradation in internal surfaces such as interarmor surfaces can be reduced by maintaining a clean, lubricated condition. On outer cable surfaces accelerated wear is usually the result of improperly selected or installed handling equipment.

2.7 Kinking - A kink results when the coil of a cable is pulled to an increasingly smaller coil diameter to the point where permanent deformation of the cable occurs. E-M cables armored with multi-layers of round metal wires are most susceptible to this condition because they usually have a tendency to rotate about the cable axis as tension increases. At high tensions, therefore, a large amount of torsional energy is stored in the cable. At low rates of tension changes this torsional energy will dissipate by counter-rotating the cable about its axis.

At high rates of tension reversals the internal friction of the cable prevents the torsional energy to be dissipated by axial rotation and coils are formed, one coil for each 360° of cable rotation.

2.8 Crushing - The crushing of an E-M cable usually occurs in situations where high compressive forces may be caused by rocks. Crushing can also occur on a winch drum when the cable is allowed to random wind and the tension coil crosses over another single coil. The high concentration of compression force can cause permanent deformation of metal strength member materials.

3.0 PARTS OF CONTRA-HELICALLY ARMORED E-M CABLES

Because over 90% of all E-M cables used in dynamic oceanographic systems use a contra-helical armor strength member, they will be discussed most completely in this chapter.

As shown in Figure 2-9, this type of E-M cable consists of two parts, the core and armor. The core consists of all components under the inner layer or armor. The armor consists usually of two layers of helically wrapped round metal wires, although 3,4 and 5 layer armors are used. The term contra-helical indicates that the layers have opposing helices.

3.1 Direction of Lay - The convention for determining right-hand and left-hand lay is the direction of the helices as they progress away from the end of the cable as viewed from either end.

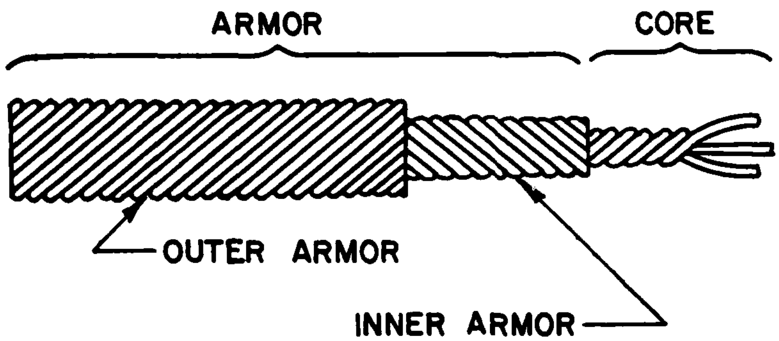


FIG. 2-9 PARTS OF AN ARMORED E-M CABLE

The cable shown in Figure 2-9 has a right-hand lay inner armor and left-hand lay outer armor. This arrangement has become an industry standard having its roots in the logging cables used in the oil industry. Because the full splicing of a cable is common practice in the oil industry, standards for armors became necessary. These standards informally developed from usage patterns of the major oil field cable users.

There is no evidence that a right-hand lay outer armor, with a left-hand lay inner armor would not provide the same performance characteristics.

3.2 Lay Angle - This is the angle the armor helix forms with the axis of the cable as illustrated in Figure 2-10. The magnitude of the lay angle is conventionally between 18% and 24%. Different lay angles are used for the inner and outer armors, the smaller angle being used in the outer armor.

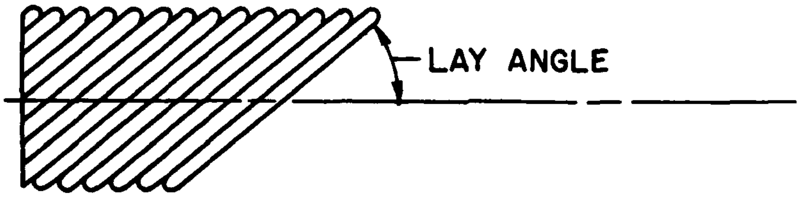


FIG.2-10 ARMOR LAY ANGLE

3.3 Preform - This is a condition of the armor wires concerning the forming of the helices during manufacturing. Before the armor wires are assembled over the underlying components (core for the inner armor and inner armor for the outer armor) they are formed into a spring-like helix.

3.4 Height of Helix - As shown in Figure 2-11, the height of the helix of the coils is determined by the internal diameter of the coil.

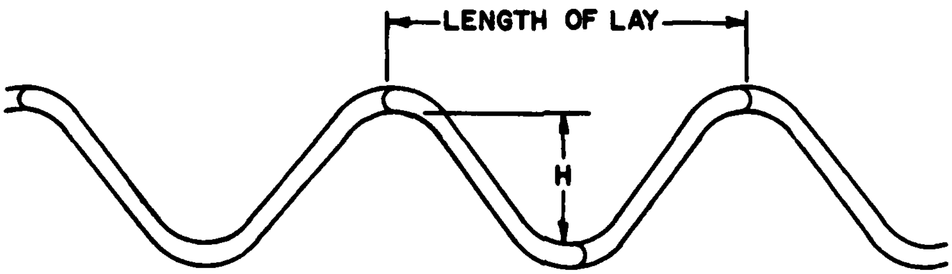


FIG. 2-11 HEIGHT OF ARMOR HELIX AND LENGTH OF LAY

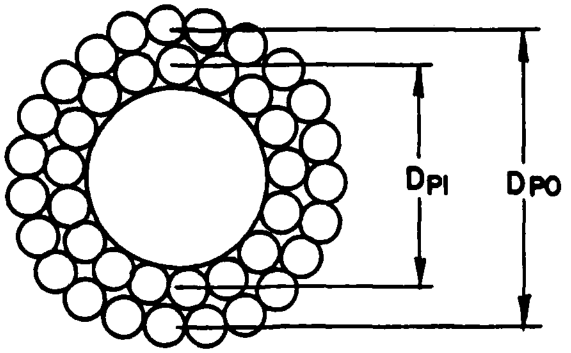


FIG. 2-12 PITCH DIAMETER

3.5 Percent Preform - The ratio for the diameter of underlying surace to the height of preform is termed the percent preform.

Example: Core dia = .240"
Height of preform = .320"

$$\% \text{ Preform} = \frac{.240"}{.320} \times 100 = 75\%$$

A 70% to 80% preform is used in current practice. Note that zero armor compression onto underlying components occurs at 100% preform; a highly undesirable condition.

3.6 Length of Lay - The length of the helix to encompass a 360° traverse is termed the length of lay. This crest-to-crest dimension is shown in Figure 2-11.

3.7 Pitch Diameter - This dimension is the diametrical distance between the center lines of the coiled wires. This dimension is illustrated in Figure 2-12 for the inner and armor wires.

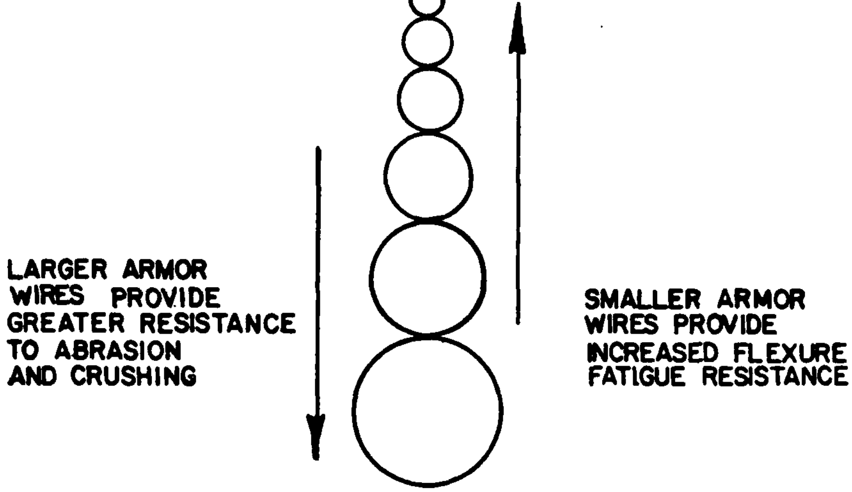


FIG.2-13 PERFORMANCE RELATIONSHIPS OF ARMOR WIRE DIAMETERS.

3.8 Number of Armor Wires - The number and diameter of armor wires are selected to cover about 97% of the surface. There is a balance between the 2-13 number and size of wires to obtain this coverage. As illustrated in Figure 2-13, for the same pitch diameter and metal the larger diameter armor wires provide greater mechanical stability; this stability relates both to resistance to distortion and to abrasion. The residual metal remaining after the same diametrical reduction by abrasion on large and small armor wires is illustrated in Figure 2-14. The percent residual metal and therefore, strength of the larger armor wires is greater.

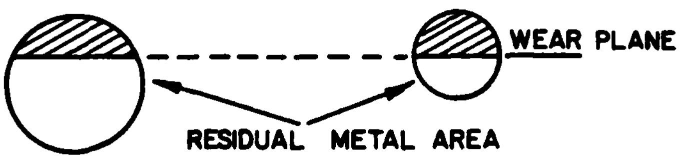


FIG. 2-14 EFFECT OF WEAR ON RED METAL AREA OF LARGE AND SMALL DIAMETER ARMOR WIRES.

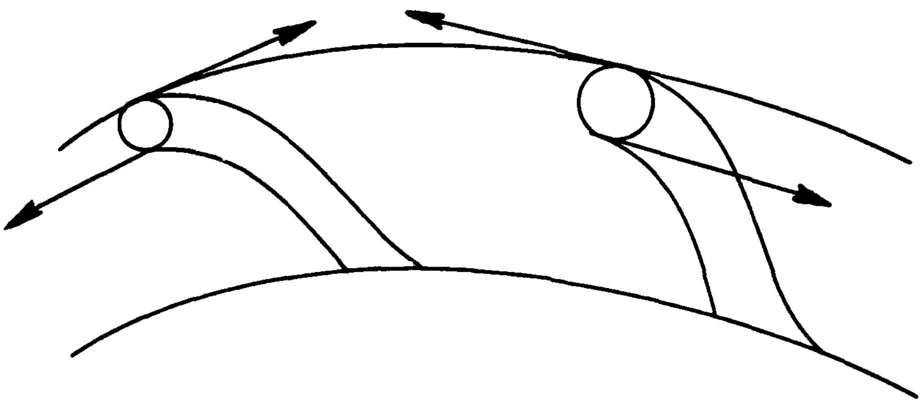


FIGURE 2-15

SMALLER OUTER FIBER STRESS IN SMALL DIAMETER ARMOR WIRES

But, for the same pitch diameter and metal, the smaller armor wires offer a greater flexure fatigue life. As illustrated in Figure 2-15, the smaller diameter armor wires will have the smaller outer fiber stress; they will, therefore, have a greater flexure fatigue life.

3.9 <u>Armor Coverage</u> - The circumference of the cable is not completely covered by the armor wires; instead, a space is allowed. This space permits greater relative movement of the individual armor wires as the cable is flexed. Also, this space permits settling of the armor layers to a smaller diameter, a natural transition for E-M cables, without overcrowding the armor wires. In a greatly overcrowded condition there will be insufficient space for all armor wires and one or more will be forced out to a large pitch diameter. In this position the wire will be higher than the others and, therefore, much more subject to snagging and increased wear; it is termed a <u>high wire</u>. A normal coverage is about 96% to 97% or omitting one wire in a 25 wire armor.

3.10 <u>Squeeze</u> - The tendency for the high compressive forces caused by the low circa 70% - 80%, preform to settle the inner armor into the core is termed squeeze. It results from the plastic deformation of the jacket or insulating surface. While much of this squeeze occurs during manufacturing and post-conditioning, it progresses during the early part of the usage period. The diametrical decrease resulting from this squeeze can be between 3% to 8%.

3.11 Core - The core may be of two general types, free-flooding or jacketed.

a. The free-flooding type of core is commonly used for oil well logging where the environment media is a mixture of oil and water at pressures which can exceed 20,000 psi. As shown in Figure 2-16, water is free to migrate through the internal parts of the core, filling the internal voids or interstices. A free-flooding cable is considered very reliable because each component is designed to be pressure-proof. Failure of one component, therefore, does not affect the function of others.

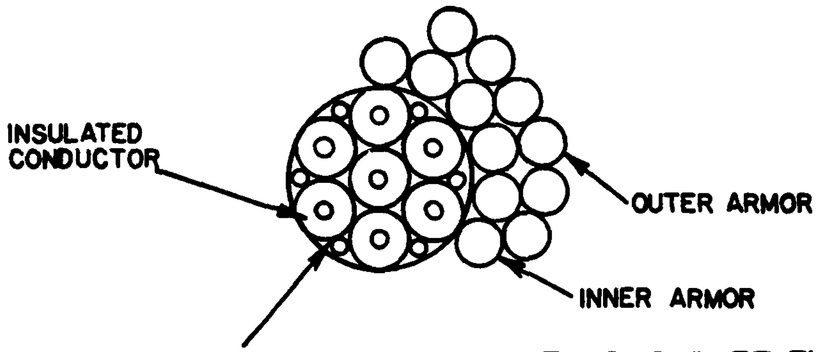


FIG. 2-16 FREE FLOODING E-M CABLE

b. In a jacketed core a pressure-proof covering is applied on the outside surface as shown in Figure 2-17. The function of the jacket is to form a pressure-proof barrier against the intrusion of water or other media into the internal parts of the core.

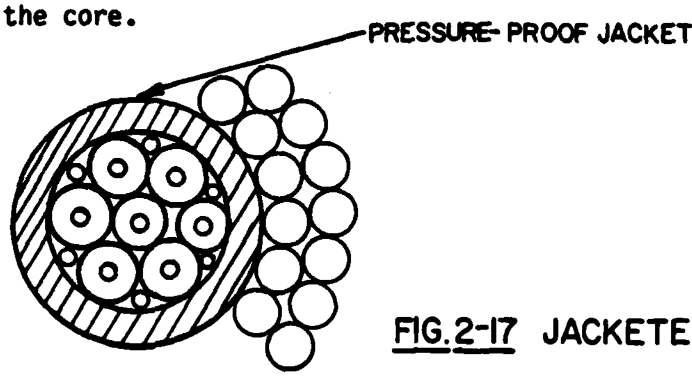


FIG.2-17 JACKETED CORE

3.12 Water Blocked Core - This term designates the type of core within which the interstitial spaces are filled with a soft material which could be depolimerized rubber, silicone rubber, vistonex or kalex. The purpose of this filling can be one of several, the primary one being the prevention of water migration axially within the core in the event of a rupture in the jacket. This filling of the interstitial voids has another benefit; it increases the compression modulus of the core as well as decreases permanent deformation termed squeeze.

Other parts of the core may also be water-blocked. The braided or served outer conductor of a coaxial core is frequently so treated as is the conductor stranding. The latter measure is infrequently used; the rationale being that cable damage severe enough to penetrate the conductor insulation has rendered in inoperable.

4.0 PERFORMANCE CHARACTERISTICS OF C-H-A, E-M CABLES

4.1 Torque Balance - This term relates to the ratio of the torque in the outer armor to that of the inner armor. Each armor unrestrained will tend to unlay; i.e., uncoil as tension is increased. The first order equation which provides a figure of merit called torque ratio (Rt) is:

$$Rt = \frac{N_O \, d_O^2 \, D_O \sin \theta_O}{N_I \, d_I^2 \, D_I \sin \theta_I}$$

where

N = number of wires per armor layer
d = armor wire diameter
D = pitch diameter of armor layer
θ = lay angle

subscripts

O = outer armor
1 = inner armor

The derivation of this equation is shown in Appendix 2. The torque ratio of most oceanographic cables is between 1.5 and 2.0. With a trade-off for other performance factors, the torque ratio can be reduced to one.

But caution must be used because:

- the above torque ratio calculation applies at one tension only; as tension is increased the magnitude of both the pitch diameter, D, and the lay angle, θ, will decrease.

- to decrease the torque ratio, Rt, a larger number of smaller diameter outer armor wires is necessary. This results in the trade-offs discussed under "Number of Armor Wires."

The effect of the number of armor wires on the armor ratio equation is illustrated in Figure 2-13. The data in the chart was taken from a selection of cables currently used in the oceanograhic applications. The expected trend toward a unity value of armor ratio as the armor wire factor increases occurs because the:

$\frac{d_o^2}{d_I^2}$ ratio becomes unity, or in extreme torque balanced cables may become less than unity.

$\frac{D_o}{D_I}$ ratio becomes very small as the diameter of armor wires (d) decreases relative to the pitch diameter (D)

$\frac{\sin \theta_o}{\sin \theta_I}$ ratio usually varies only between 0.72 and 0.83, a 15% range.

So the number of armor wires is the predominant factor in determining the armor torque characteristic.

A development of an equation expressing the torque of each armor layer and the net unbalanced torque is shown in Appendix 16.

4.2 Twist Balance - As compared with torque balance which is a potential energy function twist balance is a kinetic energy function. The tow are related in that a cable having a lower net torque can be expected to have a lower rotation vs tension characteristic. In general this is the case, but not in a direct ratio.

An important axiom to emphasize is that the two armor layers must counterrotate relative to each other for cable rotation to occur. Some factors which will decrease rotation relative to torque include:

- high armor interlayer friction
- extruded outer jacket material entering the interarmor interstices
- foreign matter entering the interarmor interstices
- a well-conditioned armor wherein the pitch diameters of both layers have reached a stable value and there is intimate contact between the inner armor and core and between the two armor layers.

4.3 Crush Resistance - This external force varies in the manner of application; it may be:

a. across one diameter as would occur by a heavy object hitting the cable while it lay on a flat surface,

b. uniform radial compression such as occurs on the underlying layers of cable spooled under tension,

c. random compressive stress such as would occur on a bottom layed cable on a shifting rocky bottom,

d. <u>self-compression</u> caused by the load end of the cable crossing over a stray loop on the drum,

e. <u>point or line contact</u> such as would occur when a cable jumped out of a sheave groove and bore the lip of the groove while under high tension.

The crush resistance of a cable increases with the use of larger diameter armor wires as depicted in Figure 2-13.

4.4 <u>Corrosion Resistance</u> - This form of armor degradation in sea water is usually associated with steel but it also occurs with various types of stainless steels. E-M cable design techniques to minimize or eliminate corrosion problems include:

a. isolation from the media by use of a covering jacket over the armor,

b. use of a corrosion-resisting metal for the armor wires,

c. (Stainless Steels Ineffective) The common types of ferritic (400 series) and austinitic (300 series) stainless steels have been found to be very ineffective for armoring materials. In addition to providing a lower ultimate tensile strength (UTS), they suffer severe pitting, referred to as crevic corrosion. This condition is aggravated by a low oxygen level in the water and is most severe in areas where there is stagnant water. Stainless steels depend on the maintenance of a self-repairing oxide coating for protection against corrosion and failure to maintain this protective coating causes severe localized metal removal by corrosion.

e. (higher alloy metals) Because of the relative low cost of galvanized improved plan steel (GIPS), the most commonly used armor metal, higher alloy stainless steels have been found cost effective in very few oceanographic cable systems. The properties of some metals which have been shown to have good corrosion-resisting properties in sea water are presented in Figure 2-18.

metal		
GIPS	10	1
Nitronic 50 (1)	4	3
AL-6X (2)	3	4

Inconel 625		2	6
MP-35N multiphase	(3)	1	10

FIG. 2-18: Comparative Cost/Corrosion of Corrosion Resisting Metals

A vital factor in the evaluation of cost-effectiveness of these higher cost alloys is the relative importance of corrosion among other cable life limiting factors such as:

- flexure fatigue

- handling damage

- abrasion

f. Factors affecting GIPS Corrosion - Because GIPS is the most commonly used armoring metal it is appropriate to examine factors which can affect the corrosion rate in sea water.

The corrosion rate of GIPS in sea water could be increased by:

- stray electric fields causing electrolysis,

- connection to system parts containing materials which are higher in the electromotive series thus rendering the steel sacrificial.

g. Decreasing GIPS Corrosion - The sea water corrosion rate could be decreased by:

- using a fresh water rinse and relubricating procedure after retrieval from salt water,

- ensuring that the steel armor is at ground potential by the proper use of grounds within the system,

- use of sacrificial zinc anodes at the terminations.

4.5 <u>Abrasion Resistance</u> - This is a metal removal degradation which can be greatly minimized by the use of proper handling equipment. Common causes of excessive abrasion include:

- improper fitting sheave grooves

- rough sheave groove surface

- cable allowed to rub against stationary surfaces

- unnecessary dragging of cable on the sea bottom.

A technique for markedly decreasing sheave groove induced abrasion in the coating of the groove surfaces with a material such as polyurethane or nylon 12.

4.6 Elongation - The percent elongation at 50% of UTS for sizes of cables which are typical to oceanographic use is listed in Appendix 17. This characteristic applies after length stabilization as described under "Prestressing" in the Manufacturing Process Section and for the same diameter of cable, will vary with:

a) core softness
b) armor tightness
c) armor construction

4.7 Sea Water Buoyancy - This buoyancy becomes more important as the immersed volume (length X Cross-sectional Area), increases. Calculation for weight in water, specific gravity, and strength to weight ratio are shown in Appendix 18.

4.8 Breaking Strength - Assuming the full conversion of armor wire strength to cable strength the cable strength becomes the sum of the strengths of the armor wires or:

$$Pc = \Sigma P_O + \Sigma P_I \qquad 1.$$

The component of armor wire tension which is parallel to the cable axis is

$$P = P\omega \cos \theta \qquad 2.$$

Where θ is the lay angle
$P\omega$ is the wire strength

The wire strength is

$$P = \frac{\pi}{4} d\omega^2 S\omega \qquad 3.$$

where:
$d\omega$ = the armor wire diameter
$W\omega$ = wire tensile strength

substitute 3. into 2.

$$P = \frac{\pi}{4} d\omega^2 S\omega \cos \theta \qquad 4.$$

Substitute Eq. 4. into Eq. 1. for the inner and outer armor wires:

$$Pc = \frac{\pi}{4} (N_o d_o^2 S_o \cos \theta_o + N_I d_I^2 S_I \cos \theta_I) \qquad 5.$$

4.81. The determination of armor wire diameter is determined by equating the circumferential length to the sum of armor wire diameters, or:

$$L = \Sigma W_C \qquad 6.$$

where L = circumferential length
ΣW_C = space occupied by wire

The circumferential space occupied by each armor wire is:

$$W_c = \frac{d}{\cos \theta} \qquad 7.$$

and the sum of all wires is:

$$\Sigma W_c = \frac{Nd}{\cos \theta} \qquad 8.$$

The circumferencial length available is:

$$L_c = \pi D \qquad 9.$$

which is decreased to allow for coverage, C(%) and:

$$L = \frac{\pi C D}{100} \qquad 10.$$

Substitute 8. and 10. in 6.

$$\frac{\pi C D}{100} = \frac{Nd}{\cos \theta} \qquad 11.$$

Solve for d:

$$d = \frac{\pi C D \cos \theta}{100 N} \qquad 12.$$

The use of Eq. 12. above to determine the diameters of the inner and outer armor wires and subsequent use in Eq. 4. will yield a relationship between <u>cable breaking strength</u> and <u>cable O.D.</u> This relationship is shown in Appendix 20.

5.0 MANUFACTURING PROCESSES FOR E-M CABLES

The processes used to manufacture E-M cables differs from those used for general industrial cables in that much greater care in quality control is mandatory. This greater attention to ensure the design integrity of components subassemblies and the final product is necessary because of the high mechanical stresses which are imposed by the armor and by the system use of the cable.

5.1 Conductor Stranding - To decrease fiber bending stresses, the elecrical conductors of E-M cables are stranded; i.e. they contain several individual wires, common STRANDINGS being 7 - 19, and 37. The lay-up is usually "bunched" which means that all wires are twisted in the same direction. Properties of copper conductors commonly used in oceanographic cables are shown in Appendix 3.

5.2 Insulation - The majority of electrical insulating materials are thermoplastics with the most commonly used being ethylene propylene copolymer (polypropylene), polyethylene, and fluoropolymers (Teflons).* These thermoplastic materials are supplied in granular pellet form. These pellets are put into an extruder which melts them and feeds the melt to the extruder head where the semi-liquid thermoplastic is formed around the conductor wire as it traverses through the extruder die. The coated wire is then cooled in a long trough filled with flowing water.

Tests which are usually conducted at this stage are insulation diameter and electrical integrity by means of a spark test.

a.) Diameter measurements are electro-optically made in two orthogonal planes by electro-optical instruments, laser based instruments being popularly adopted.

b.) The spark test consists of elecrically stressing the insulation by a voltage generally in the region of 8,000 to 14,000 volts. The purpose is to induce an insulation breakdown where a weakness may occur. These weak insulating paints may be caused by voids (bubbles), inclusions (foreign mateiral) or extreme non-uniformity of the wall thickness (non-concentricity).

5.3 Wet Test - All insulated conductors are subjected to another electrical test while submerged in water. The reel containing the completed conductor is fully immersed, except for the ends, in fresh water to which a chemical (wetting agent) has been added to lower surface tension and thereby improve wetting of all the insulation surface. After soaking for a period of up to six hours, electical tests are made of:

* Tademark of DuPont

- dielectric strength (hipot)

- insulation resistance (IR)

The insulation resistance valves range above tens of thousands of megohms.

5.4 Cabling - In this process several conductors are twisted together either to form a group which may, in turn, be further cabled with other groups to form the final electrical part of the E-M cable core.

5.5 Braiding - When the electrical core is a coaxial conductor, the outer conductor shield may be braided which is the same construction used on coaxial cables specified in MIL-C-17.

5.6 Serving - Because of the self-cutting tendency of braided copper outer conductors at the wire crossover paints served shields have become popularly used. This construction consists of helically wrapping several wires around the insulation in the same manner as armor wires are applied. The trade-off is that a lower percent coverage in the region of 80% to 85% is the maximum which can be expeced with a served outer conductor. To increase coverage a metal or metal-coated mylar tape is sometimes used.

5.7 Jacketing - The same extrusion process as used for insulating is also used to apply the jacket over the core and/or the armor. To test the pressure-proof in water integrity of jacketed constructions, tank testing is sometimes used. Because of the limited availability of pressure test tanks of sufficient volume and because of the high cost, these tests are usually omitted.

a.) One technique sometimes used to increase the reliability of a jacket invovles the use of a double layer extrusion. This procedure greatly reduces the chance of a pin-hole, bubble or other flaw in one layer from being coincident with a similar minor defect in the outer layer.

b.) Reinforced Jacket - When a two-layer jacket is used there is an opportunity to greatly increase its tensile strength by using an open braid of a high strength fiber over the first extrusion. Candidate materials include polyester or aramid yarns. This jacket construction is called a reinforced jacket.

c.) Common thermoplastics used for jackets include:

- polyethylene (high density, low molecular weight)
- polyurethane (polyether)
- hytrel
- nylon

d.) When specifying or selecting a jacket material foresight should be given to the termination procedure. If a potted termination is to be used, the bonding procedure should be established.

5.8 Armoring - When the electrical core is completed it is installed into the armoring machine and the spools of armor wires are loaded into cradles of the armoring machine. Two general types of armoring machines are in common use; 1) the tubular type and 2) the planetary type. Both types are in successful use for producing high quality armors.

a.) In recent years the tubular armoring machines have become more frequently used because of production efficiency. They operate at up to 2,000 rpm as compared with a usual maximum of 300 rpm for planetary machines.

b.) A major consideration during the armoring process is to have sufficient lengths of wire in each spool to make the entire cable. This avoids planned welds in the armor wire. User specifications frequently limit the number of welds in the outer armor layer and specify the minimum distance between welds along the cable.

c.) An example specification control of welds is that:

- minimum distance betwen welds shall be 5,000 ft.
- no more than one weld in any one armor wire
- no more than three welds in each armor layer.

d.) Armor Wire Welds - Broken armor wires are usually butt fusion welded. The heat of welding anneals the metal in the vicinity of the weld and vaporizes the galvanize coating. The primary function of the weld is to provide a smooth mechanical tansition across the broken section. Therefore, the loss of approximately 50% of the unwelded wire strength has a minor effect on the performance capability of the cable.

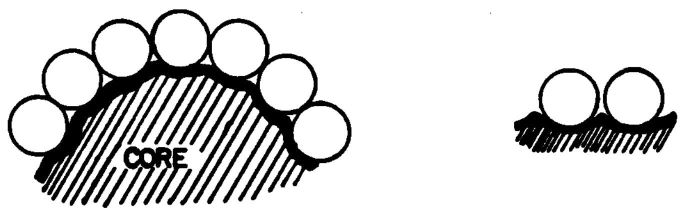
5.9 Prestressing - is a term applied to the stabilizing of the construction of an E-M cable; it is also termed length stabilization. When first manufactured, the inner armor wires seat into the underlying thermoplastic insulation or jacket as shown in Figure 2-19. This is an unstable condition because of the very high surface stress which at working loads can exceed the yield strength of thermoplastic.

Manufacturers, therefore, will prestretch the cable by passing it over several sheaves at a tension of about 40% of the breaking strength. The equipment used for this operation, called prestressing, varies but functionally includes equipment shown in Figure 2-20.'

The objective of this operation is to operationally stabilize the cable; i.e., reach a condition wherein the same

FIG. 2-19 GEOMETRY CHANGES DURING CABLE CONDITIONING

2-19A AS MANUFACTURED THERE IS A SMALL INDENTATION OF INNER ARMOR WIRES INTO THE CORE MATERIAL.



2-19B WHEN LENGTH STABILIZED, OR PRESTRESSED, THE INNER ARMOR WIRE CONTACT WITH THE CORE INCREASES.

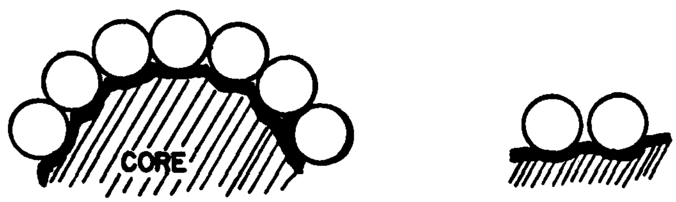
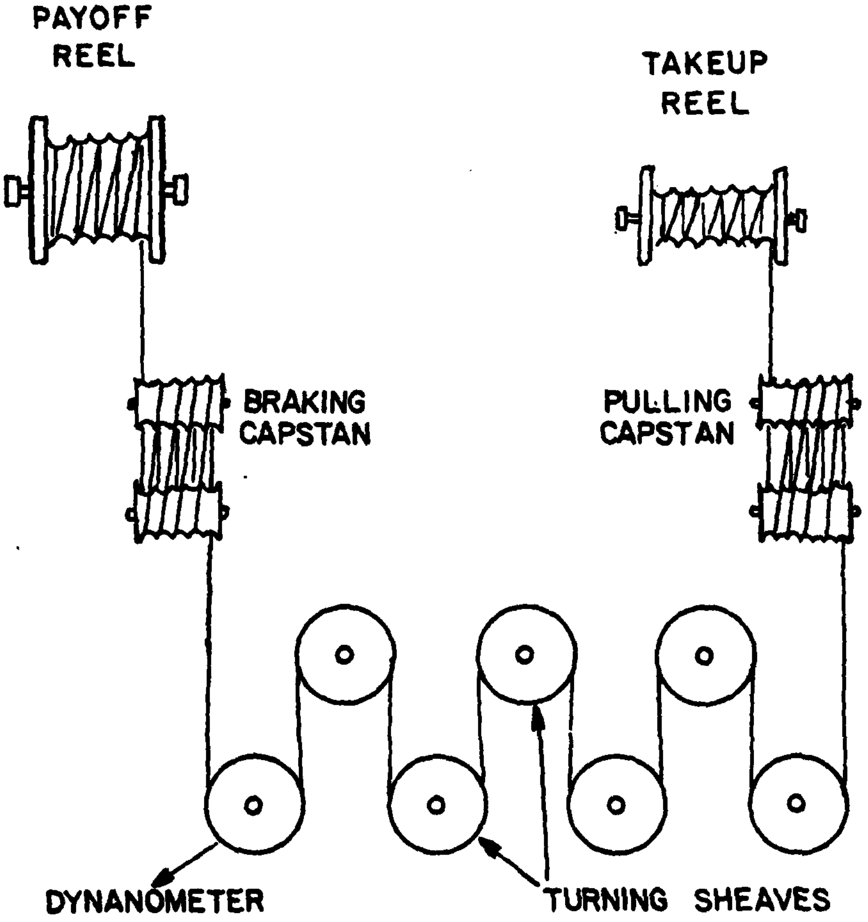
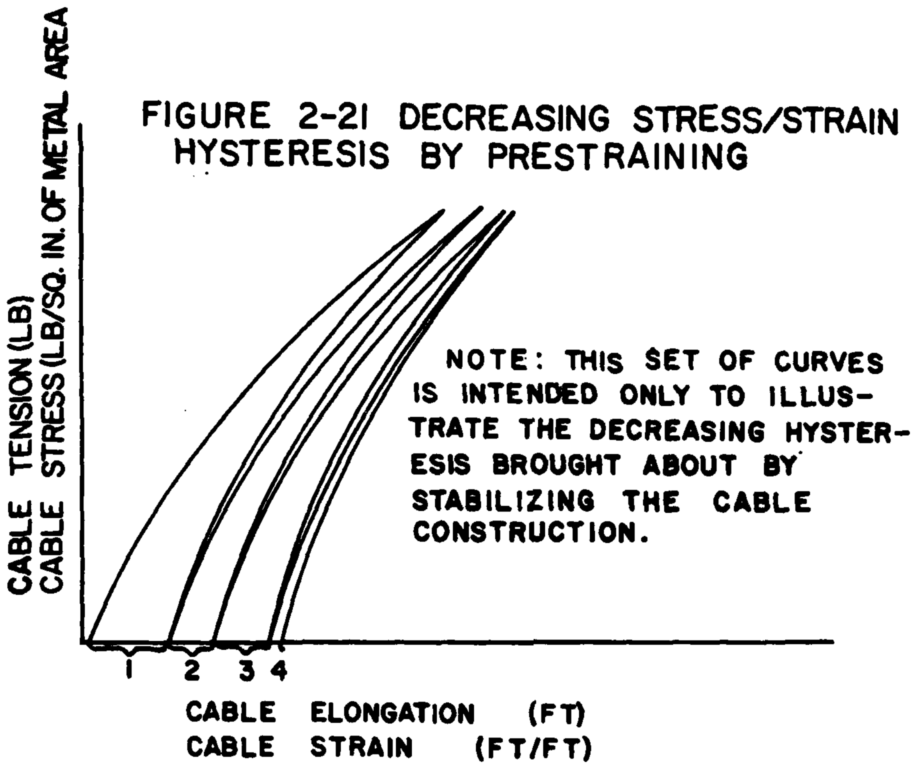


FIGURE 2-20
PRESTRESSING EQUIPMENT



cable squeeze and consequent cable stretch will occur on subsquent tensioning operations. This hysteresis phonomena graphed in Figure 2-19 is decreased as the inner armor contact with the core increased as illustrated in Figure 2-19. This indentation will continue until a contact surface is formed, which results in a stable contact stress valve.

FIGURE 2-21 DECREASING STRESS/STRAIN HYSTERESIS BY PRESTRAINING



6.0 Handling E-M Cables

The discussion of handling E-M Cables starts with the assumptions that the cable had been properly specified and procured.

6.1 <u>Storage Before Use</u> - E-M cables are usually supplied on heavy duty steel or wood shipping reels. The cable will be uniformly thread-layed on the reel; i.e., it will be tightly coiled with no gaps or crossovers. This practice is to prevent intransit damage to the cable which can occur due to self-crushing at these crossovers.

The reel should always be stored upright; i.e., resting on the two flanges. Storing the reel on the flat of one flange can cause coils to cross over into a random tangle. Subsequent righting of the reel and proper re-reeling of the cable can be very difficult. When stored in an unsheltered area, the reels

should be covered with the bottom left open for ventilation. If the storage period is to be more than a few months, the spraying or wiping of an extra amount of lubricant onto the surface layer of cable will provide added protection.

The reel should be lifted by using a bar through the center holes. In no case should fork lift blades bear onto the coiled cable.

6.2 Spooling Effect on E-M Cables - The spooling of a cable onto a storage drum can be performed with only sufficient tension to tightly pack the cable in a thread-lay. A tension of 3% of the cable breaking strength is reasonable.

When using a single drum winching system the winching power and storage function are provided by a single unit and the installation of the cable becomes critical. Before discussing spooling procedures, the effects on the cable should be noted. A contra-helically armored E-M cable is most resistant to damage by compressive forces when all components, i.e., two armor layers and core, are intimately in contact so that there is little relative movement from the external force. When these cable parts act as a unit, the resistance to damage by compression is maximized. Also, when the compressive force is distributed around the cable circumference rather than across one diameter, less distortion will result in a much lower damage possibility.

TABLE 1

Typical Spooling Tension Schedules

Cable Dia. in	Cable Dia. mm	Approximate UTS lbf.	Spooling Tensions (lbf) first layer	second layer	third layer
.187	4.8	3,600	360	430	540
.203	5.2	4,500	450	540	680
.219	5.6	5,200	500	600	750
.250	6.4	5,900	600	700	900
.297	7.5	6,700	670	800	1,000
.312	7.9	10,300	1,030	1,240	1,550
.350	8.0	9,200	920	1,100	1,380
.375	9.5	13,900	1,400	1,670	2,090
.438	11.1	17,900	1,800	2,150	2,700
.469	11.9	17,000	1,700	2,050	2,550

For succeeding layers the tension in the third layer is maintained for about one-half the total cable length.

For the remaining half of the total cable length the tension is reduced in equal increments every 1,000 ft to the first layer value at the outer layer of cable.

6.3 Smooth Drum Spooling - Smooth drum spooling uses a plain cylindrical winch drum and is most commonly used on small oceanographic winches containing 1,000 to 2,000 meters of cable. Because of the low deployment forces invovoled, the spooling onto these winches is less critical. However, good practice dictates that a uniform thread-lay be used.

6.4 Tension Spooling Objectives - When longer cables are to be handled by a tension winch formalized spooling procedures become mandatory to prevent cable damage. The procedure has three objectives:

a. Tightly thread-lay the cable under tension to ensure that the cable cross-section has resistance.

b. Provide sufficient rigidity of cable in lower layers to prevent nestling or keyseating of the tension coil.

c. Provide sufficient spooled tension to balance some of the deployment tensions to reduce coil slippage caused by tightening of the tension coil.

6.5 Tensions for Spooling - Spooling tensions vary in succeeding layers according to schedules which vary according to the experience of many able technicians. The schedules shown in Figure 19 applies for a selection of small diameter cables. The tensions shown in Table 1 are typical for E-M cables similar to those used in oil well work. They will vary for different types of armor.

The schedule shown in Figure 2-22 displaying spooling tensions expressed as a percentage of the cable UTS, or breaking strength, is applicable to a wide variety of cable types.

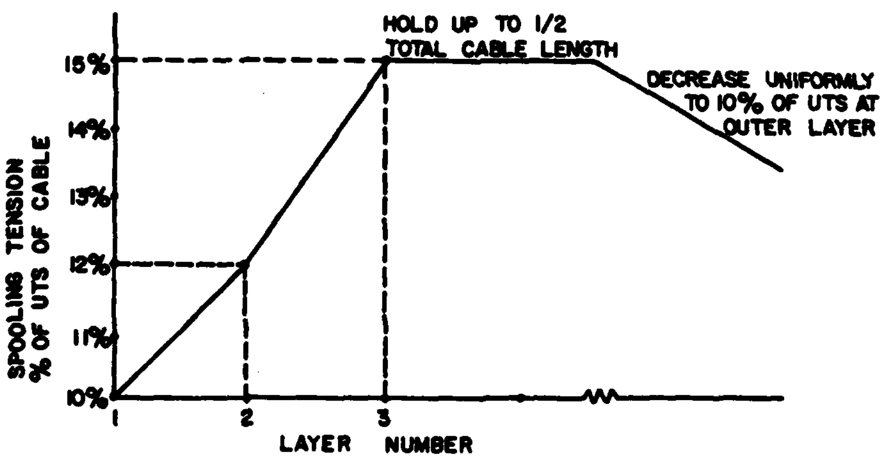


FIG. 2-22 SPOOLING TENSION SCHEDULE

6.6 Lower Spooling Tensions - It is very possible to obtain satisfactory performance using much lower spooling tensions with the provisions:

a. Care is taken that the bed (first) layer is properly started using a sufficiently high spooling tension with coils in tight contact and uniformly distributed across the winch drum.

b. The remaining spooling operation is performed with good workmanship at tensions as close to the recommended valves as possible.

c. MOST IMPORTANT± Initial deployments are made at low tensions and slow winching speeds. The tensions can be gradually increased in subsequent deployments.

6.7 Grooved Drum Sleeves - These grooved drum sleeves, made by Lebus, Inc., are described in a later chapter. Their use is encouraged because they:

a. determine the spooling thread-lay at the bed layer and, therefore, more positively ensure that the remaining procedure will be correct.

b. similar to correct sheave grooves, these grooved sleeves provide support for the cable to increase its crush resistance.

6.8 Sheaves - The correct sheave groove design as illustrated in Figure 2-23 provides for circumferential cable contact of about 40° of arc.

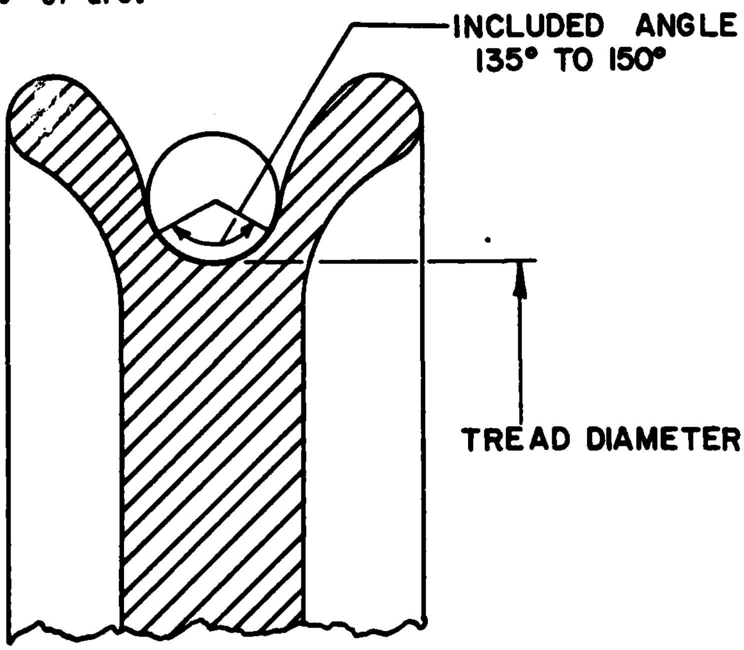


FIG.2-23 CORRECT SHEAVE GROOVE DESIGN.

a. Groove surface finish: The surface of the groove should be smooth with a surface roughness not exceeding 32 microinches. As grooves become worn, the surfaces will become corrugated and cause accelerated abrasive wear of the cable; they should then be refinished.

b. Hardness of groove surface: The hardness of the groove surface should be less than that of the armor wire which is about Rockwell (C-scale) 55. It is less costly to resurface a sheave groove than to replace an E-M cable. Good results have been obtained with the use of polyurethane coated sheave grooves; other thermoplastics have also been successfully used.

c. Tread diameter: The tread diameter of the sheave should be as large as possible, the minimum diameter for a cable following the general rule:

$$d - 400\ dw$$

where

D = sheave tread dia (see Figure 2-23)

dw = largest diameter of armor wires in the cable

d. Bending Stresses: The importance of designing an E-M cable handling system to impose a minimum number of directional changes in the cable can be understood by an examination of bending stresses. To allow a cable to bend, the inner and outer armor layers must move relative to each other. This causes abrasion between the armor layer; i.e., the importance of lubrication. Also, the shortening of the lay angle at the entrance point of tangency and normalizing at the exit point of tangency causes a rubbing action between the cable and sheave. The wear rate resulting from this abrasion will increase because of increased

- cable tension
- winching speed,

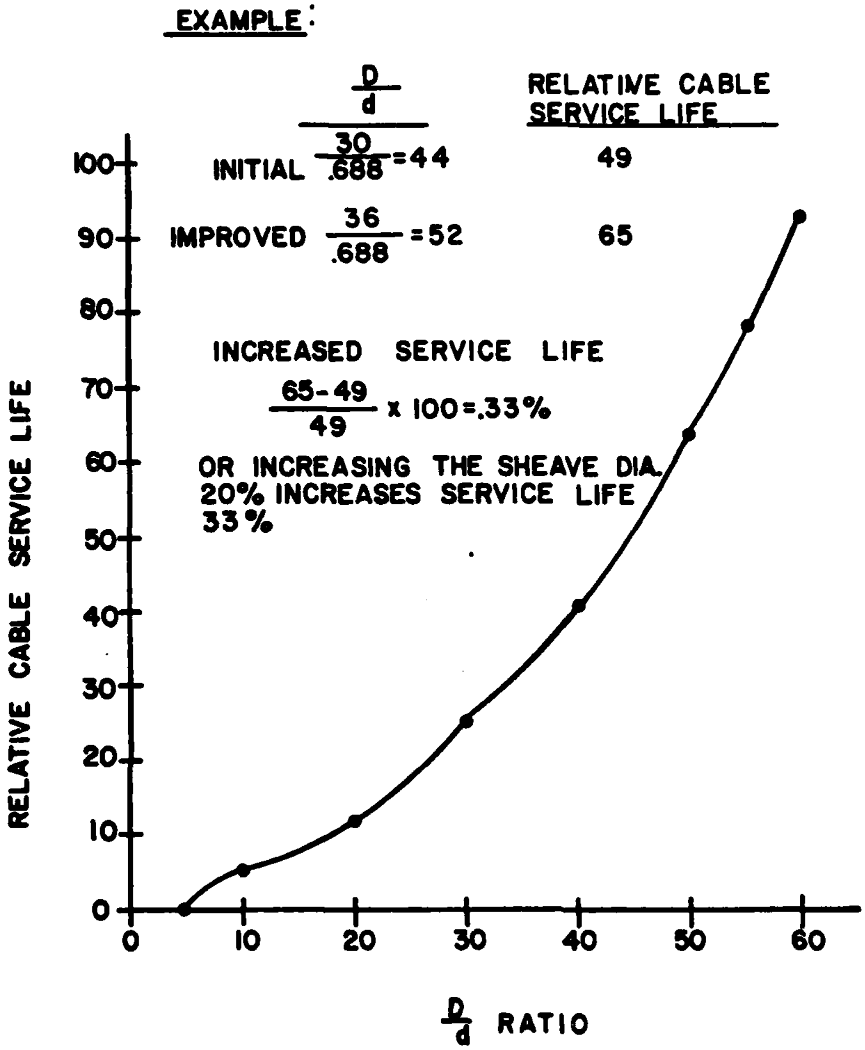
and a decreased ratio of

$$\frac{D}{d} = \frac{\text{sheave tread diameter}}{\text{cable diameter}}$$

e. Sheave tread diameter effect on flexure fatigue life: Another illustration of the importance of sheave tread diameter to the flexure fatigue life is presented in Figure 2-24 and 2-24a.

7.0 Field Inspection and Testing

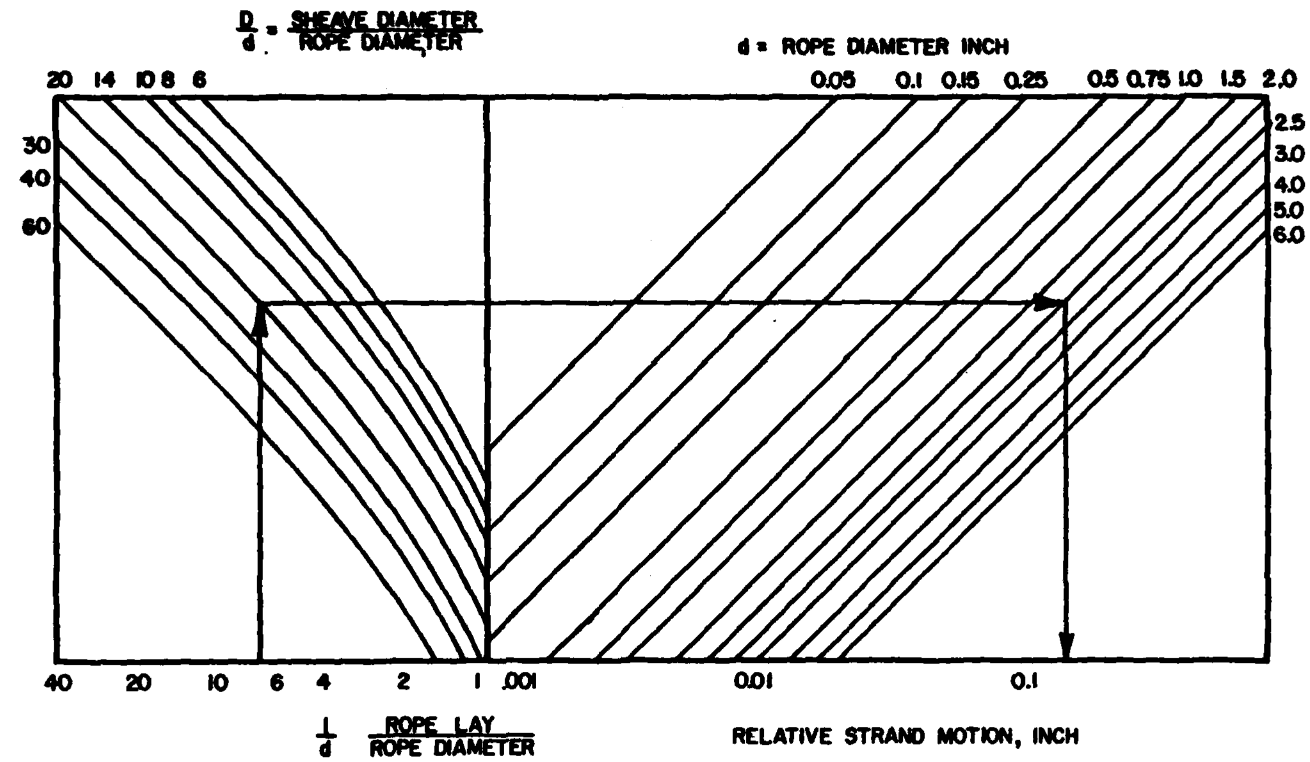
7.1 General - When an E-M cable has been properly specified (see the next section of this chapter) the inspections advisable upon receiving it essentially concern verification that the receiving records conform to the purchase specifications.



REF : WIRE ROPE USERS MANUAL, 1981
AMERICAN IRON AND STEEL INSTITUTE

FIG. 2-24 RELATIVE SERVICE LIFE FOR VARIOUS $\frac{D}{d}$ = RATIOS

FIG. 2—24a INFLUENCE OF ROPE AND SHEAVE GEOMETRY ON RELATIVE STRAND MOTION FOR A SIX STRAND CONSTRUCTION



7.2 Required Inspections - After a cable has been put into service, there is need to use inspection procedures for:

a. location and identification of performance defects which may occur because of cable handling or service oriented accident.

b. monitoring of changes in cable geometry and performance characteristics caused by usage wear. Information obtainable from this data can prove to be very valuable in deciding to retire the cable from service.

7.3 Cable Record Book - As is common practice on oil well electrical wireline (oil well logging) trucks which use E-M cables similar to those used for oceanoraphic instrumentation, a cable record book is highly recommended for the oceanographic community. It is much more necessary for cables used on oceanographic survey ships because of the rotation of personnel; operating engineers on oil well electrical wireline trucks may use the same E-M cable for its entire life. He, therefore, will know of any special performance characteristics and of the usage history.

Data appropriate for the Cable Record Book and other data arrangements may be found more applicable to certain systems. The important consideration is that a record of cable usage is maintained. The cost effectiveness of E-M cables and the relative performance lines of similar cables in similar systems otherwise cannot be accurately evaluated.

7.4 Cable Log - This parallel record to the record book is a history of the status of cable characteristics and of maintenance and repair prcedures.

7.5 Inspection - An E-M Cable should be inspected and undergo maintenance after each sailing or at regular intervals of time. The inspections should include:

a. visual to observe, evaluate, and document any damage or severe wear and log its location in the Cable Log;

b. armor tightness which can serve as a warning that the cable was overstressed or that the cable should have Service Shop maintenance. This latter activity is described in a later section of this chapter;

d. conductor electrical resistance, which can be another indicator of overstressing. Also, it can indicate conductor damage from other causes. In any event, this is valuable data for determining the suitability of the cable for continued use.

e. outside diameter, like outer armor lay length is an indication of length stability. But, when visual examination shows abrasion of the outer surface of outer armor wire, it is

an indication of residual metal cross-section and, therefore, are residual UTS.

f. need for lubrication, before storage a cable should always be lubricated. Materials and procedures are discussed in anothr chapter.

7.6 Visual Inspection Practice - This is performed while respooling to or from the winch to or from a storage reel at a slow, less than 50 ft/min speed. Observe for any changes in the armor which can include:

a. permanent offset, which is an indication that there may have been the onset of a kink or that the cable was overstressed by bending over a very small diameter while under a tension in excess of 10% of UTS;

b. scraped or nicked wires, as evidenced by tneir being at a larger diameter than the others. This is indicative also of these wires experiencing shear forces caused by the cable being drawn over stationary surfaces. It may also indicate an overtension experience.

7.7 Armor Tightness Inspection - Three methods are possible for this evaluation; they are:

a. pick test, which is the easiest but the most qualitative cf the three is illustrated in Figure 2-25. If an outer armor wire can be raised above the outside diameter of other outer armor wires, a loose armor condition is indicated. The same test should be performed at several locations to determine if any loose armor indication may be localized.

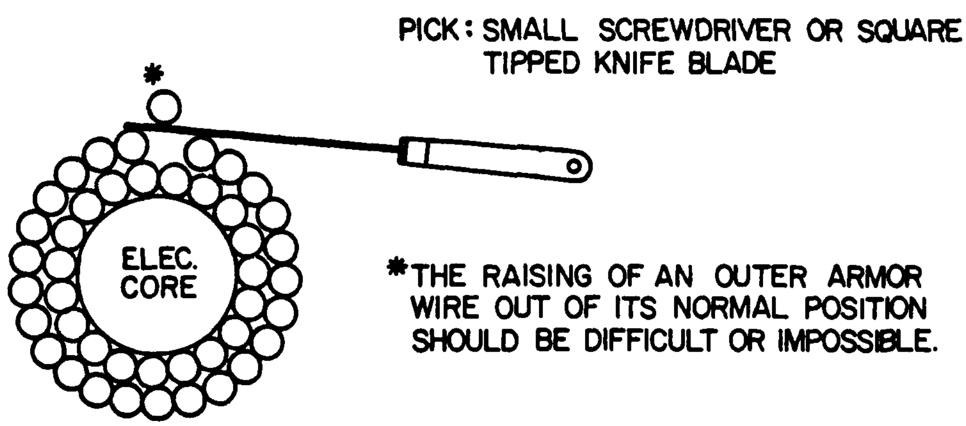
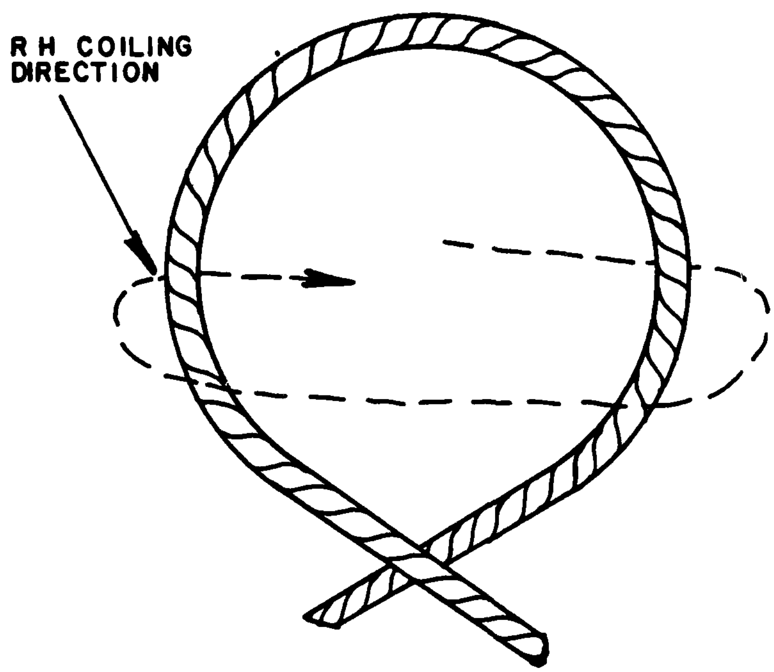


FIG. 2-25 PICK TEST FOR LOOSE ARMOR DETERMINATION

FIG. 2-26 LOOP TEST FOR LOOSE ARMOR DETERMINATION (FOR L-H-L OUTER ARMOR)

1. OPERATOR STANDS BESIDE A RUN OF SLACK CABLE WHICH IS BY THE RIGHT HAND.
2. WITH THE PALM OF THE RIGHT HAND FACING AWAY FROM THE BODY, THUMB TO THE REAR, THE OPERATOR GRASPS THE CABLE.
3. THE HAND IS TURNED CLOCK-WISE 180° OR UNTIL THE THUMB IS POINTING FORWARD, THUS FORMING A COIL AS DIAGRAMMED BELOW.
4. IF THERE IS A LOOSE OUTER ARMOR THE CABLE WILL REMAIN COILED OR TEND TO CONTINUE FORMING ADDITIONAL COILS.



b. coil test, which is performed on a run of cable by forming an in-line coil having a coiling direction which tends to tighten (axially rotate the cable in a direction opposite the outer armor lay direction) the outer armor. For the standardized L-H-L outer armor the coil direction becomes R H. The coil test procedure is described in Figure 2-26.

c. catenary test, which is performed on a run of cable by farming a catenary on a 30 ft. to 40 ft. run. When the catenary is aligned with a vertical reference such as a plumb bob, a deviation of the catenary in the direction tending to tighten (see above) the outer armor indicates a loose outer armor condition. For a L-H-L outer armor a loose armor condition will be indicated by a deviation to the right of vertical; the larger deviations indicating a higner severity of looseness. This inspection procedure is diagrammed in Figure 2-27.

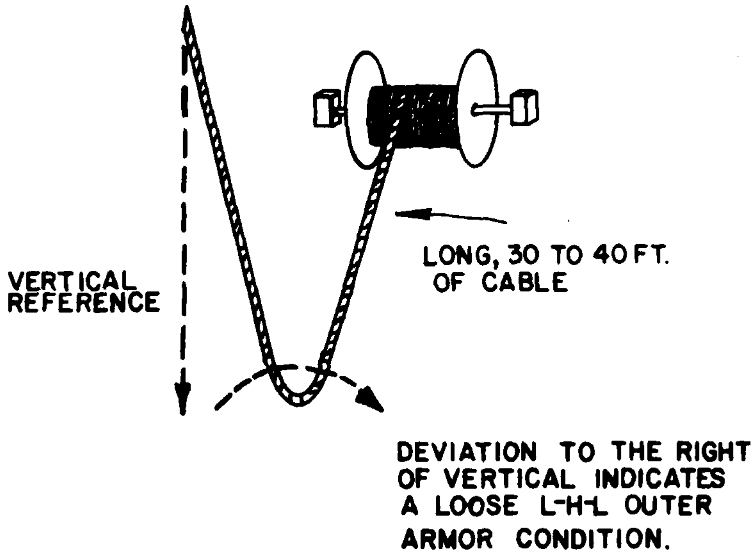
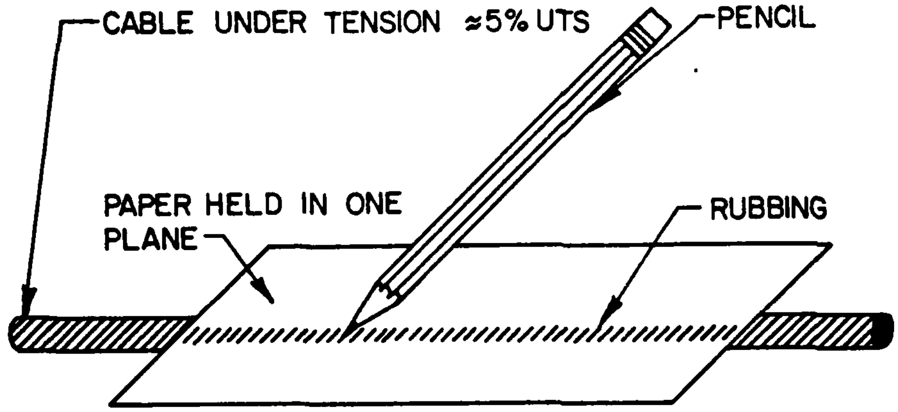


FIG. 2-27 CATENARY TEST TO EVALUATE LOOSENESS IN L-H-L OUTER ARMOR.

7.8 Lay Length of the Outer Armor

a. The general lay length of a single armor wire helix is shown in Figure 11. This lay length will increase as the cable becomes length stabilized, a natural occurrance in service. The progression of this lay length increase can be used as an indicator of our armor wire loosening or of cable core deterioration. It is therefore, one of the cable's vital signs.



1. LAY PAPER OVER A STRAIGHT RUN OF CABLE.

2. RUB THE PENCIL ON THE PAPER ABOVE THE CABLE WHILE THE PAPER IS HELD IN ONE PLANE.

3. THE RUBBING SHOULD SHOW A MARK FOR EVERY COIL OF THE WIRES. THE MARKS SHOULD BE APPROXIMATELY THE SAME LENGTH.

AN ALTERNATE PROCEDURE INVOLVES INKING THE CABLE AND PRESSING IT BETWEEN TWO BOARDS ON WHICH THE PAPER IS PLACED BETWEEN CABLE AND BOARD.

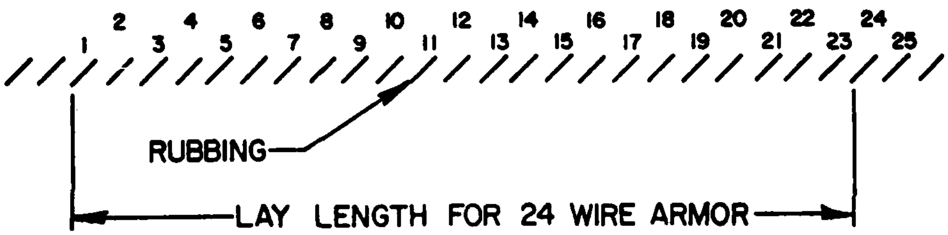


FIG. 2–28 MEASURING PROCEDURE TO DETERMINE ARMOR LAY LENGTH

b. Procedure, the procedure for field determination of lay length is shown in Figure 2-28 to obtain accuracy, the rubbing from the cable must be in one plane and the markings of uniform length; the paper must not be allowed to rotate while the rubbing is being made. Also, remember that when a cable is held in a curved position, the lay length on the outside surface of the curve has a longer lay length. Therefore, it is important to perform this inspection procedure on a straight run of cable.

The rubbing is measured by laying a high accuracy scale on the rubbing and reading the scale with optical magnification. Remember, the lay length will vary with tension; therefore, the rubbing should always be taken on the cable while under the same light tension, about 5% of UTS.

7.9 Conductor Electrical Resistance

a. Procedure: Changes in the conductor electrical resistance may indicate that damage has occurred and an evaluation must be made of the continued usefulness of the cable.

The desirable instrument for this test is a resistance bridge having an accuracy of 0.1%. The temperature of the conductor has a significant bearing on resistance so that the cable must be kept in a shaded, near constant temperature area for a minimum of twelve hours prior to taking this measuremnt to ensure that the entire length is at a uniform temperature.

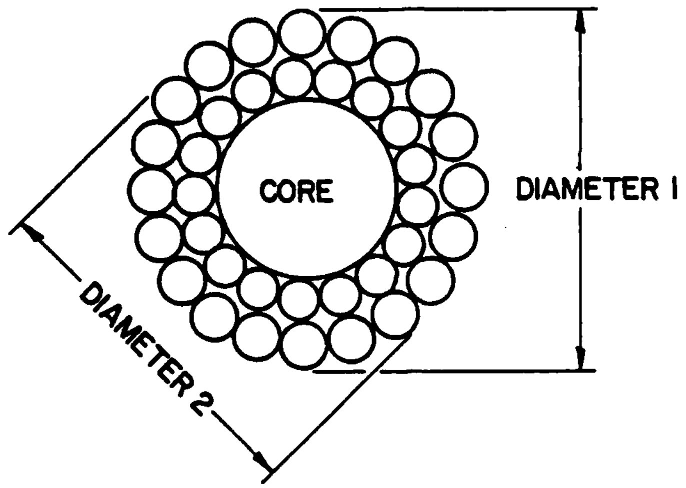
The standard temperature for expressing electrical resistance is 20°C (68°F); to convert the electrical resistance taken at any other temperature to the valve at 20°C, use the correction factors shown in Appendix 4.

b. Cabled conductor resistance: Remember that the measured electrical resistance is for the conductor in the cable and will apply directly only if the conductor is coincident or parallel, with the cable axis. Otherwise, the readings must be corrected for cabling. Appendix 5 shows the correction which must be made to express electrical resistance as cabled to straight wire electric resistance. When the wire is axially coincident with the cable $\theta = 0$ and $\tan \theta = 1$.

7.10 Outside Diameter - As for measuring the lay length of the outside armor wires, the cable must be maintained under a known, repeatable, constant tension when measuring the diameter. A reasonable value of this tension is about 5% of UTS.

a. Caliper method: The micrometer or vernier caliper method of diameter measurement is shown in Figure 2-29. Note that the two orthogonal measurements are a minimum. When a large variation (greater than 1%) occurs between the two measurements, others should be taken to locate the largest

diameter. The measurements are averaged to obtain an average diameter.



MEASURE DIAMETERS ACCROSS WIRES ON TWO ORTHOGONAL AXES AND AVERAGE.

FIG. 2-29 MEASURING DIAMETER USING A MICROMETER OR VERNIER CALIPERS

b. Measuring tapes: Measuring tapes with the measurement of circumference divided by . These tapes are available from manufacturers of measuring tapes and the ones used for diameters less than one inch have 0.002 inch graduations. These tapes are useful because they provide an average diameter directly (see Figure 2-30).

7.11 Need for Lubrication - An inspection to determine the need for lubrication is based more on judgment than on any measurable property of the cable.

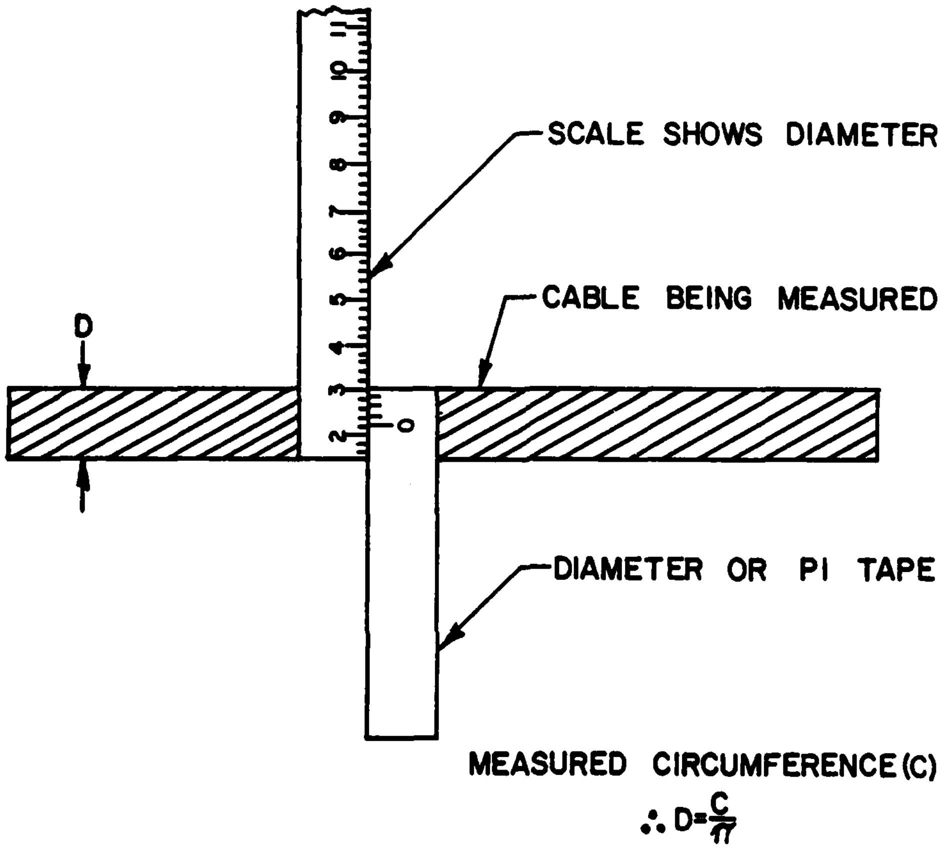


FIG. 2-30 DIAMETER TAPE METHOD OF MEASURING DIAMETER

a. Opening the Outer Armor: The most direct method would involve using clamps such as vise grips and axially twisting the cable in a direction to open the outer armor. For a L-H-L out armor this invovles imposing a R-H rotation to the R-H vise grip as shown in Figure 2-31.

Figure 2-31

This procedure should initially be tried on a piece of scrap cable to learn how much the outer armor can be displaced without causing any permanent deformation.

When the outer armor is opened, observations should be made of the presence of lubricant.

b. Inter-armor layer wear: While the outer armor is opened, also observe the extent of inter-armor wear; i.e., wear at the armor wire crossovers (see Figure 2-32). The rate of this wear is greatly affected by the maintenance of lubrication. Other factors affecting this rate include:

1.) bearing pressure over sheaves (see Appendix 3)
2.) winching speed
3.) presence of abrasive materials

c. Bearing pressure determination: The bearing pressure parameter is described in Appendix 6 where the maximum allowable value for wire rope use with cast carbon steel sheaves is 1,800 lbf/sq. in. The calculated values of bearing pressure for typical oceanographic instrumentation E-M cables as shown in Appendix 7 are less than 200 lbf/sq. in. The reasons for such low values is the much lower strength-to-diameter ratio of E-M cables compared with wire rope and the common use of a 5:1 safety factor for E-M cables in oceanographic systems.

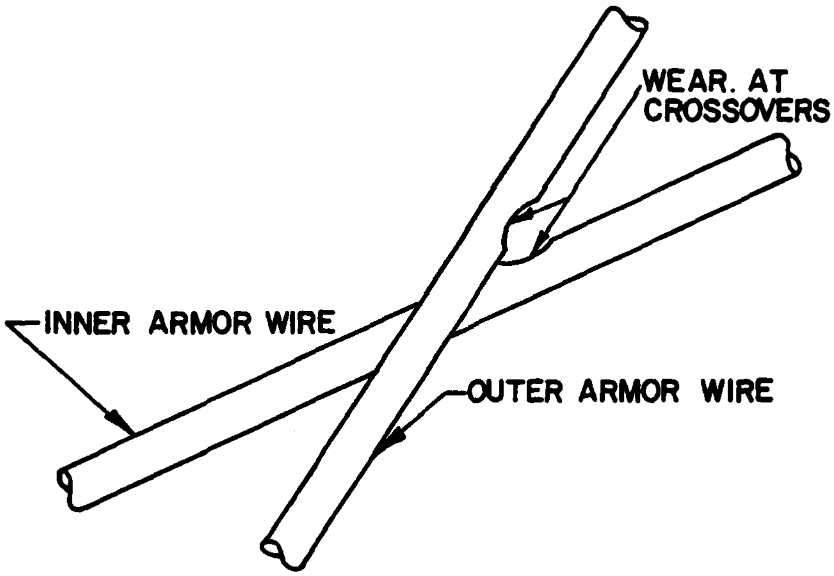


FIG. 2-32 ARMOR WIRE WEAR AT CROSSOVERS

1.) Measure the capacitance of conductors adjacent to the faulty conductor, record C_a and C_b.

2.) Average C_a and C_b

$$C_{avg} = \frac{C_a + C_b}{2}$$

3.) Measure C_1 and C_2 and locate the open by:

$$L_1 = \frac{C_1}{C_{avg}} L \qquad L_2 = \frac{C_2}{C_{avg}} L$$

4.) A time domain reflectometer (TDR) may be used to locate an open in a conductor of a single or multi-conductor cable. The proper use of this instrument requires calibration on a length of the same cable to determine the dielectric constant. When used with multi-conductor cables, corrections for the lay angle must be incorporated.

7.12 Location of an <u>open</u> in a conductor -

1a. For a single conductor cable

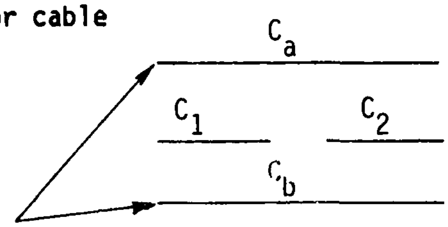
1b. Using a capacitance bridge measure the capacitance from both ends of the cable; C_1 and C_2.

1c. The length to the open is determined by:

$$L_1 = \frac{C_1}{C} \qquad L_2 = \frac{C_2}{C}$$

C = capacitance in pf/ft from manufacturers data

2a. For a multi-conductor cable



Conductors adjacent to the conductor having an open.

1.) Measure the capacitance of conductors adjacent to the faulty conductor, record C_a and C_b.

2.) Average C_a and C_b

3.) Measure C_1 and C_2 and locate the open by:

4.) A time domain reflectometer (TDR) may be used to locate an open in a conductor of a single or multi-conductor cable. The proper use of this instrument requires calibration on a length of the same cable to determine the dielectric constant. When used with multi-conductor cables, corrections for the lay angle must be incorporated.

7.13 <u>Fault location</u>, conductor short - The short may be conductor-to-conductor or, most commonly, conductor-to-armor. The detection of a high resistance short is extremely difficult to locate and, in most cases, the residual insulation resistance must be reduced to a direct short by applying a high voltage for burning through the remaining insulation. These procedures require specialized equipment and skills and should be performed only by adequately experienced personnel. Service Centers as maintained by manufacturers of oil field electrical wirelines offer this and other services as further discussed in Section 12. The modified Murray Loop Test which is applicable, is described in Appendix 14.

7.14 <u>Re-Reeling</u> - The setup for performing many of the inspections in this section is diagrammed in Appendix 13. Motive

power can be provided by any means which permits the operator to start and stop easily. Although well equipped cable service centers use variable speed, reversible hydraulic drives field inspection/repair stations have successfully used electric motors and:

a. friction drive against the edge of the reel flange,
b. V-belt

7.15 Cable length determination - The original length of an E-M cable will reduce as service continues. This reduction can be caused by the normal wear factors or by handling damage. The measurement of the length of a long cable can be determined by:

1. a footage marker tape. This is a continuously marked tape which is installed in the cable during manufacturing. See Section 10 for specification coverage.

2. re-reeling,

3. conductor resistance,

4. weight.

b. The footage marker tape offers the most convenient method of determining the length of an E-M cable. This tape is installed in the cable by the manufacturer and it is marked with sequential footage figures. To determine the length of a cable it is necessary to read the footage numbers at each end and subtract. The specification of the footage marker tape for having it included in a cable is covered in section 10.

c. Re-Reeling is convenient for length determination when it is being performed for inspecting the cable. Otherwise it is a very time consuming and difficult method.

d. Conductor resistance offers an accurate and convenient means for length determination; it requires a high accuracy resistance bridge. The procedure for single and multi-conductor cables is shown in Appendix 14.

e. Weight of the cable offers an approximation of cable length, but is the least accurate of all methods. The procedure is described in Appendix 15.

8.0 RETIREMENT CRITERIA

8.1 Considerations - Optimum, known reliability in use is the objective of the several activities bearing on the cable during the conception to retirement (cradle to grave). These activities ideally encompass:

a. systems analysis to establish a full set of requirements (Chap. 7)

b. using these requirements to draft a cable procurement specification (Chap. 2)

c. verification of conformance of the cable by review of manufacturer's test reports and conducting Receiving Inspections (Chap. 2)

d. proper design of the handling system (Chap. 4, 8, 9, 10, 12)

e. proper installation in the cable system (Chap. 2)

f. proper operation of the cable system (Chap. 2)

g. proper cable maintenance (Chap. 2, 3, 5).

g. evaluation against established criteria to determine fitness of the cable for continued use.

8.2 . Broken Wire Criteria - The wire rope retirement criteria established by the American National Standards Institute and discussed in Chapter 1 are not applicable to E-M cables. The major percentage of total cable strength of E-M cables is contributed by the outer armor which is composed of single wire, not multi-wire, strands, armor wires. The breaking of any one of these armor wires is of major concern becasue it is subject to unlimited unstranding unlaying. This unstranding relates to the ability of the broken wire to become continuously unwound from the cable.

Therefore, when an outer armor wire of an E-M cable becomes broken, the first action is to determine the cause. These causes together with follow-on activities include:

a. external abrasion: local or general. The usual cause is rubbing against a stationary surface or roughened sheave grooves. Should other wires appear serviceable, the broken wire may be repaired and service resumed.

b. broken factory weld: in this case an inspection of other facatory welds which may be in the same cable is indicated. If the remainder of the cable appears satisfactory, the broken weld may be repaired and the cable returned to service.

c. nicking by a sharp object: as for the broken weld above, the remainder of the cable should be carefully inspected to determine if other nicks exist. If the damage is not extensive, a decision could be made to repair the broken wire and other nicked wires and return the cable to service.

d. broken wire in a crushed cable section: very careful electrical measurements are necessary to determine circuit integrity. If the cable is electrically satisfactory and the other wires in the damaged area are satisfacatry, the decision could be made to return the cable to service.

e. wear between armor layers: a broken wire attributed to this cause indicates a general deterioration of the cable. The remainder of the cable should be carefully examined to determine the extent of this wear by opening the outer armor at measured intervals along the cable, not more than 1,000 ft. or 20% of total length, whichever is greater.

f. corrosion: this cause is difficult to distinguish from "E," above, because corrosion usually causes an acceleration of wear between armor layers as shown in Figure 32. The same inspection procedure as in "E" should be performed and if a decision is made to repair the broken wire and resume service, the cable should be carefully cleaned and lubricated.

g. kink: this massive localized deformation is cause for immediate removal from service, or splicing of usable lengths (Appendix 19).

h. birdcaging: this is a localized evidence of a general improper operational condition. A birdcage is caused by a sudden release of tension whereby the potential energy of cable stretch induces an axial compressive strain causing permanent deformation of the wires. As for a kink, this condition is cause for immediate removal from service. The electrical core will usually have been damaged.

8.3 Life Cycle Criteria - a. Repetitive Cable Usage Systems: In systems which make repetitive use of the same cable, the reliability requirements may be so high that periodic replacement whether on service, mission life or cycle life may be imposed. Both of these criteria demand the maintenance of accurate records. When these criteria are used, much information usable for modifing the retirement criteria are obtainable from the used cables.

b. Cable log information usage: The log of these used cables, together with final inspection reports, form a valuable information bank not only for use in modifying retirement criteria, but for use in identifying the life limiting factors. These factors can lead to investigations to improvements in the areas of:

1. E-M cable design concept
2. E-M cable engineering design
3. handling system design
4. handling procedures
5. maintenance procedures

c. Resulting increase in service life: and therefore, the collected data would be used to increase the service life and cost effectiveness of the E-M cable.

8.4 Non Destructive Testing - a. History: In the period of 1975 to 1982, there has been a steady development of techniques for detecting anomalies in steel wire constructions. They are generally based on ultrasonic electromagnetic and Hall effect phenomena.

b. The ultasonic method of anomaly detection and identification was the standard technique used in Project THEMIS, which was conducted in the period of 1972 to 1979, at the Catholic Unviersity of America, Washington, D.C. An ultrasonic transducer was connected to one end of a wire rope specimen being tested for UTS and pickup mounted on the other end. The change in transmission through the wire rope specimen provided warning that changes in the structure were occurring. By comparing the recording of the nature of changes in ultrasonic transmission to the observed structure changes, a set of standards were developed which permitted accurate failure prediction later in the program.

c. Hall effect: Instruments using the Hall effect principle are becoming popularly used at this writing and within the Navy (NSRDC - Annapolis) work is in progress to develop a system for ultimate Navy operational use. A Hall effect instrument is commercially available from a company in the Netherlands. That instrument has been approved by Del Norski Veritas and Lloyds of London for certification of ropes used in aerial tramway systems across the Alps. The largest mining company in Canada, Noranda, Ltd., has developed a Hall-effect instrument for inspecting and certifying wire ropes used in their mines. Two companies presently offer a service for inspecting steel wire structues.

d. Development of retirement criteria: That these instruments and services for using them are becoming available is encouraging for establishing retirement criteria based on the change of metal area and construction characteristics throughout the cross-section and the entire length of cables.

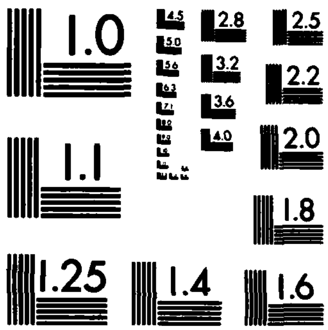
9.0 CABLE MATERIALS

9.1 Conductors - The most frequently used conductor material is copper because of its high conductivity and reasonable price. This is the only material to be considered in this section.

a. Circular mils: The area of round conductors is expressed either in circular mils (CM) or square millimeters (mm^2). A circular mil is the diameter of a circle (expressed in mils) squared.

AD-A132 513 HANDBOOK OF OCEANOGRAPHIC WINCH WIRE AND CABLE TECHNOLOGY(U) RHODE ISLANDUUNIV KINGSTON GRADUATE SCHOOL OF OCEANOGRAPHY A H DRISCOLL 18 OCT 82 2/5

UNCLASSIFIED N00014-82-G-0116 F/G 8/10 NL



MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

From Appendix 3 the seven wire strand of a No 20 AWG conductor is seven wires, each .0126" dia. The diameter of each wire in mils is 12.6 and the circular mils of each wire is 158.76. For the seven wires, the total circular mil area is 7 x 158.76 = 1111 as shown in Appendix 3.

b. Gaging system: The U.S. standard is American Wire Gage (AWG). This system is organized on the basis of defining two wire sizes and basing all others on those sizes. The two defined sizes are:

4/0 diameter = 0.4600 in.
36 diameter = 0.0050 in.

There are 38 sizes (39 increments) between these base points. Therefore, the ratio of diameters of adjoining sizes is:

$$\sqrt[39]{\frac{0.4600}{0.0050}} = \sqrt[39]{92} = 1.29$$

An approximation for estimating the relative properties of wires of various AWG gage numbers is:

- an increase of three gage numbers doubles the area and weight and halves the electrical resistance.

c. Coating: The major purpose for coating copper to prevent oxidation. The use of tin, silver or nickel depends on expected temperatures, the limites being:

tin	135° C	(275° F)
silver	200° C	(392° F)
nickel	300° C	(572° F)

Because of the near exclusive use of non-corroding thermoplastics for conductors, insulations in oceanographic E-M cables bare copper is generally used for the conductor material.

c. Phydical Properties: The physical properties of annealed copper are:

specific gravity	8.89
tensile strength, psi	35,000
elongation at 10% of UTS	20

e. Stranding: All conductors used in E-M cables are of a standard construction, i.e., they contain several individual wires which are twisted into a composite. As shown in Appendix 3, the number of wires can be varied, generally 7 and 19-wire for conductors up to Δ4 AWG. A larger number of wires can be used where even greater flexibility and tolerance to bending

fatigue is required. The method used for twisting wires of a conductor is called stranding; the common construction being "bunched" or twisted together so that all have the same lay. The lay length of strands is generally eight times the strand diameter.

Common specifications for copper conductors are ASTM-STD-B286, specification for copper conductors for use in hook-up wire for electronic equipment.

9.2 Electrical Insulations - With very few exceptions, the insulations used in oceanographic E-M cables are thermoplastic. As the name implies, this class of plastics have a repeatable relationship of physical properties with temperature. The properties of the most commonly used materials are shown in Appendix 8.

The most commonly used insulating material, polypropylene, has the lowest specific gravity and very good electrical and mechanical properties, equivalent, or better than those of polyethylenes and fluorocarbons.

The next most commonly used insulating material is polyethylene; it is the standard material for the insulation of coaxial cables.

The government specification covering insulated conductors is MIL-W-16878, and coax cables are covered by MIL-C-17.

9.3 Shielding - This part of a cable is usually the outer conductor of a coax, but can also be an electromagnetic interference shield of a single or multiconductor component.

The material for shields may either be tapes or a construction of round wires in either a braid or a serve. Because of the tendency of tapes to break in small E-M cables, their use is usually limited to large multiconductor cables.

a. Shielding tapes commonly used are a mylar base with a film of copper on one side. To provide electrical continuity of wraps and a means for termination, a drain wire is used. This drain wire is cabled with the conductor bundle and lies in one of the outer interstices. The size and location of the drain wire must provide for this electrical contact with the conducting surface of the shielding tape. Shielding tapes have the advantage of low cost, and 100% shielding; but the disadvantage of poor mechanical properties, particularly those required in E-M cables for flexing service.

b. Braids utilize small diameter copper wires in sizes generally within Δ30 AWG to Δ38 AWG. The coverage is between 85% and 95% and it is the highest cost shielding method.

Its use in flexing E-M cables should be adopted with caution because of widespread experience with extreme degradation. The nature of this degradation is self-cutting of the wires at the crossover points. The very high compressive forces of the covering contrahelical armor imposes extremely high stresses at these point contacts.

c. Serves use small diameter copper wires as in braids, but the difference lies in the construction. A served shield is like the serving of the armor in E-M cables; it may be single layer or double layer. The percent coverage for a single serve is lower than that for braids being in the range of 80% to 90%. A contrahelical served shield may provide a coverage approximating that of a braid. The advantage of a served shield is a longer flexure fatigue life than either the taped or braided shields. The higher flexure fatigue life compared with braids results form elimination of the high stress.

Although not substantiated, the practice of filling the voids in served shields may additionally increase the flexure fatigue life. This filling may be silicone rubber, vistonex, or other suitable material.

9.4 <u>Jackets</u> - Two types of jackets must be considered in an E-M cable, that which is under the armor and that which covers the armor. The requirements for the physical characteristics of the materials do not differ extremely; they may be summarized as:

- low water permeability
- low cold flow characteristic
- high abrasion resistance
- high cut-through resistance
- resistance of petroleum compounds

Three of the compounds included in Appendix 8 (HDPE, polyurethane, Nylon 6) generally satisfy these requirements. TPR is greatly affected by petroleum compounds and has poor abrasion and cut-through resistance; its primary attribute being the low, 0.88, specific gravity.

The thickness of jackets for the usual diameter range of E-M cables approximately follows the 10% rule; i.e., the normal jacket thickness is 10% of its inner diameter.

9.5 <u>Armor</u> - The most common metal used for armor is steel because of its relative low material cost, excellent mechanical properties, and ease of fabrication and assembly.

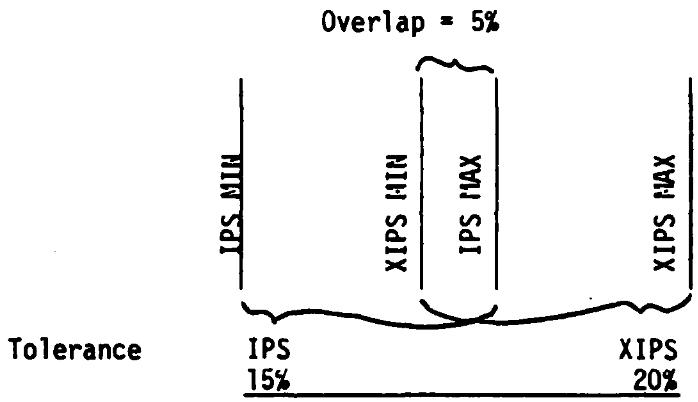
a. Steel grades, the steel wires for E-M cables, are covered by the same AISI (American Iron and Steel Institute) specification which applies to wire rope. The grades covered in this specification are:

- mild plow steel
- plow steel
- improved plow steel
- extra-improved plow steel

The tensile strength increases to the highest in extra-improved plow steel.

The two grades commonly used in E-M cables are improved and extra-improved plow steel. The breaking and tensile stengths for a selection of wire sizes normally used in oceanographic cables is shown in Appendix 9.

The relationship between the maximum and minimum tolerance values should be noted; they are:



When quoting breaking strenghs, manufacturers usually state a minimum value which, for wire they may purchase, could have the above tolerances. The E-M cable manufacturer, having a wire mill, has the opportunity to tailer the wire and, therefore, more idealize the strength-to-diameter characteristics. A standard AISI test for ductibility of these wires is to wrap the wire in a close helix for six complete turns around a mandrel having a diameter twice that of the wire being tested. There should be no tendency to develop cracks or to break.

The Government specification for these steel wires is: RR-W-410, "Wire Rope and Strand."

b. Stainless steels: The austinetic (300 Series) stainless steels, particularly Type 316, has been used in oceanographic cables with no success in obtaining a longer life by eliminating corrosion as the life limiting operational factor. Corrosion experienced by steel was found much less hazardous than the insidious crevice corrosion to which this class of stainless steels are susceptible.

bl. Crevice Corrosion - Because stainless steels depend on the maintenance of a protective oxide film to isolate the basis metal from seawater oxygen starvation can expose the

highly reactive basis metal. In double (contra-helical armor construction there is little water flow into the inter-armor area and oxygen depletion occurs. The consequent breakdown of the oxide film allows localized corrosion which is termed crevice corrosion.

b2. High Alloy Steels - Two alloys, Inconel 625 and MP-35N have been very successfully used in highly corrosive environments. Their properties are shown in Appendix 10. Although their use has been very successful with no history of difficulties, the very high cost has discouraged any more than highly specialized use.

b3. Nitronic 50* and AL-6X** - Both of these proprietary alloys are being used in current government systems. Nitronic 50 armored cables are being used in cables for Navy Tow systems and at this writing fleet evaluation is still in progress. AL-6X was extensively tested before being selected as the armoring metal for the OTEC power cables. Properties are shown in Appendix 10.

With the tradeoffs in lower tensile strength and higher cost, both alloys appear to offer a longer service period before corrosion becomes the basis for retirement. The economic study for each system must be based on the effet of corrosion as being the major life-limiting factor. Also, in many cases, the higher cost of these corrosion resisting alloys must be balanced with the cost effectiveness of an improved cable maintenance program.

10.0 CONTRA-HELICALLY ARMORED, E-M CABLE SPECIFICATION

The development of a meaningful cable specification requires a thorough analysis of system equipment and phenomena which affect the operation of the cable. Because of the uniqueness of each system generalized specifications do not help either the user or the manufacturer. The approach in this section will be the discussion of information which should be considered in the development of a TAILORED procurement specification.

10.1 Performance vs Construction Specification - A construction specification is the most simple tool for communication of requirements to the manufacturer. This approach is based on the presumption that either the same, exact cable construction provided satsifactory performance in the same or in a similar system.

10.2 Construction Specification - When using a construction specification all known restraints should be explicitly stated. For instance, to fit an established Lebus grooving or conform to

* Nitronic 50: Trademark of Armco Steel
** AL-6X: Trademark of Allegheny Ludlum Steel Co.

some other handling system restraint, the cable feature or system description should be stated.

a. A convenient technique is to use a manufacturer's part number for the description. It must be remembered that changes may have been made in the design of the cable and the statement should either refer to a particular procurement or date in addition to the part number.

b. Important data to include in a Construction Specification are shown in Appendix 11. Note especially the request for test data to be furnished to the Buyer. These data are valuable to permit any later trouble shooting the cable should it become damaged in service.

10.3 Performance Specification - This is the most useful type for the system designer because it requires less knowledge of cable design, but more knowledge of performance requirements. Also, this approach does not restrict the manufacturer in the effort to provide the best design for the system.

As shown in Appendix 12, the major specification elements are:

a.) Scope
b.) Referenced Documents
c.) Requirements
d.) Test, or quality control
e.) Marking and Shipping,

and each section will be discussed further.

a. Scope: The type of system and operational information will, together with the life objective provide the cable designer with valuable information regarding tolerance to tensile or flexure fatigue and mechanical trauma.

b. Referenced Documents: Care must be taken in referencing only those documents which are used in desribing the Requirements, Section 3. The statement used in Government Specifications is "The following documents are a part to this specification to the extent specified herein." The last statement, "to the extent specified herein," requires that there is specific reference to a particular part of the document and the only purpose for listing the document is for the convenience of the reader.

c. Requirements: This section contains the details of requirements which should extend to the requirements impacting the electrical and manufacturing termination

d. Test: Care is needed to include all testing which is necessary to ensure that critical performance characteristics are covered in the qualification test program and verified in

production tests.

e. Marking and Shipping: length - The cable may be marked by:

- by a footage marker tape
- by magnetic marking

The footage marker tape is a thin Mylar tape which is Layed longitudinally along the cable core. It is marked in feet and permits a very convenient determination of residual length and positively discriminates one end from the other. Also, by recording the footage at each end upon receiving the cable, the amount of cable cut from either or both ends can be determined as the cable continues in service. The cost of these footage marker tapes is extremely low and they are easily installed during manufacture.

Magnetic marking is a technique of permanently magnetizing points on steel armor. It is a standard technique for oil well logging cables wherein these marks are used for measuring the length of payed-out cable. The readout instruments are commercially available and all manufacturers of oil well logging cables have the magnetic marking equipment. Unlike paint or tape markers, these magnetic marks are not removed by abrasion or reaction with sea water.

Consideration should have been given to handling of the shipping reel so that any restrictions to the diameter or width dimensions should be stated.

11.0 AVAILABLE CABLE SERVICES

11.1 General - Just as has the origin of E-M cables occurred in oil industry service for logging, perforating, etc., service centers have also become available from the same industry. E-M cables in the oil field services receive much more continuity of use than in oceanographic services. Therefore they experience wear phenomena which is partially reversable by proper services.

These services are tailored to the needs of a standardized range of oil field cables which have an upper diameter of about 1/2 in. Many oceanographic cables are included in this range and some services may be extended to encompass other diameters.

It is useful to know of the availability of these services and in general the procedures used to obtain maximum, satisfactory service life.

The simplified splicing steps presented in Appendix 19 are intended only to show the principles which govern this

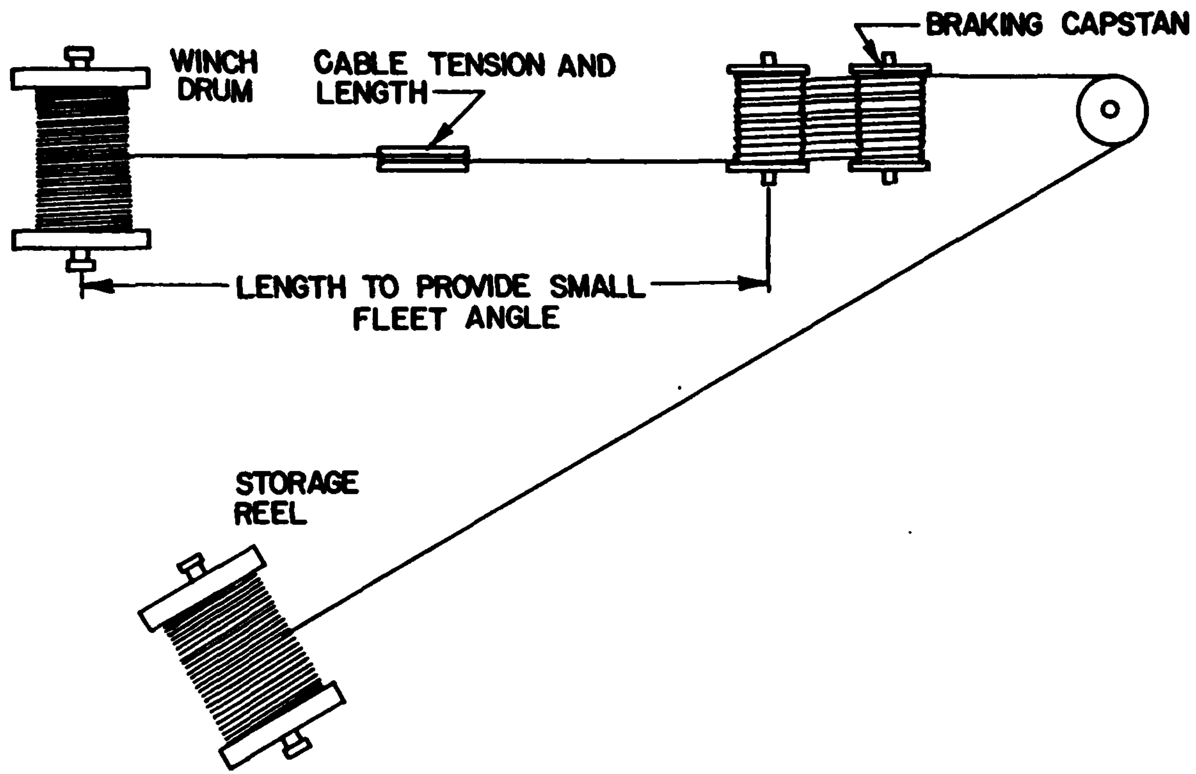


FIG. 2-33 SPOOLING SETUP

process.

11.2 Spooling - To obtain an optimized spooling setup specialized equipment and skills are required. Although some oceanographic fitting-out facilities have adequate equipment, many do not.

The elements of a spooling setup used in Service Shops are shown in Figure 2-33. The Braking Capstan provides a regulated back-tension on the cable to obtain the spooling schedule discussed in Section 6.0 and in Chapter 10.

11.3 Splicing - A full cable splice including all core components and the armor is possible; and is routinely performed on oil field cables. This is one of the E-M cable oils which depends on apprenticeship learning. Very little is published and, outside the oil field cable community, it is relatively unkown. Principles which apply to an E-M cable splicing procedure are presented in Appendix 19.

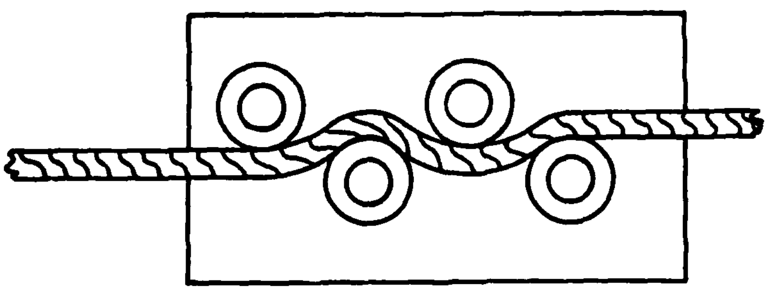
11.4 Fault Location - All equipment and skills required for fault location are available at service shops. In general, all inspection and testing discussed in Section 7.0 can be performed at Service Centers.

11.5 Reconditioning - Reconditioning is a series of operations performed on a used cable to effect:

(a) cleaning
(b) retightening the outer armor
(c) relubrication

a. Cleaning is accomplished by rotary wire brushing the external surfaces. When foreign materials are lodged in the inter-armor area dislodging is encouraged by passing the cable through offset rolls.

FIGURE 2-34 OFFSET ROLLS



DUAL SETS ARE PLACED IN ORTHORGONAL AXES

b. Outer armor tightening is performed by mounting the reel of cable in a reel turner. The amount of tightening is evaluated by one of the methods described in Section 7.7 or, in some cases observing the tendency of the cable to rotate about its axis as a point translates from the lower turning sheave to the tensioning device (capstan or hoist).

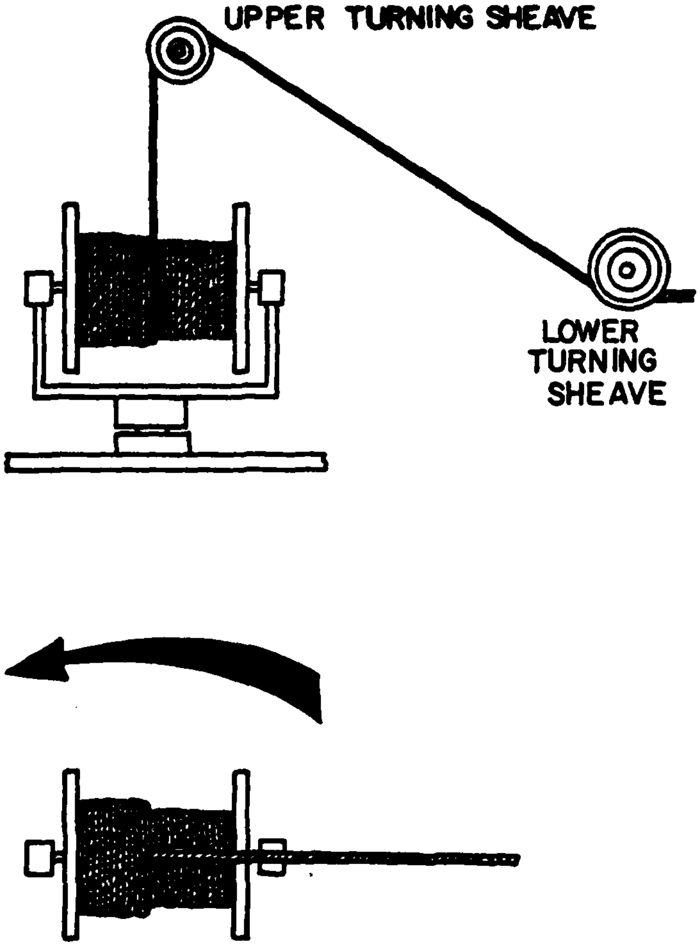
Rotation in a LHL outer armor tightening direction (near end rotates CW) indicates looseness, and a need for additional rotations of the reel turner per 100 feet of cable.

c. Lubrication is performed in a pressurized tube which is fitted with end glands to seal around the cable. The general schematic of a pressure lubricator is illustrated in Figure 2-36.

11.6 Magnetic Marking - A means for reliable, accurate cable pay-out determination, is performed by appling a high level magnetic flux to a localized part of the armor. The measurement between magnetic marks has been standardized at 100 feet and these length increments are determined automatically by referencing to the previous mark.

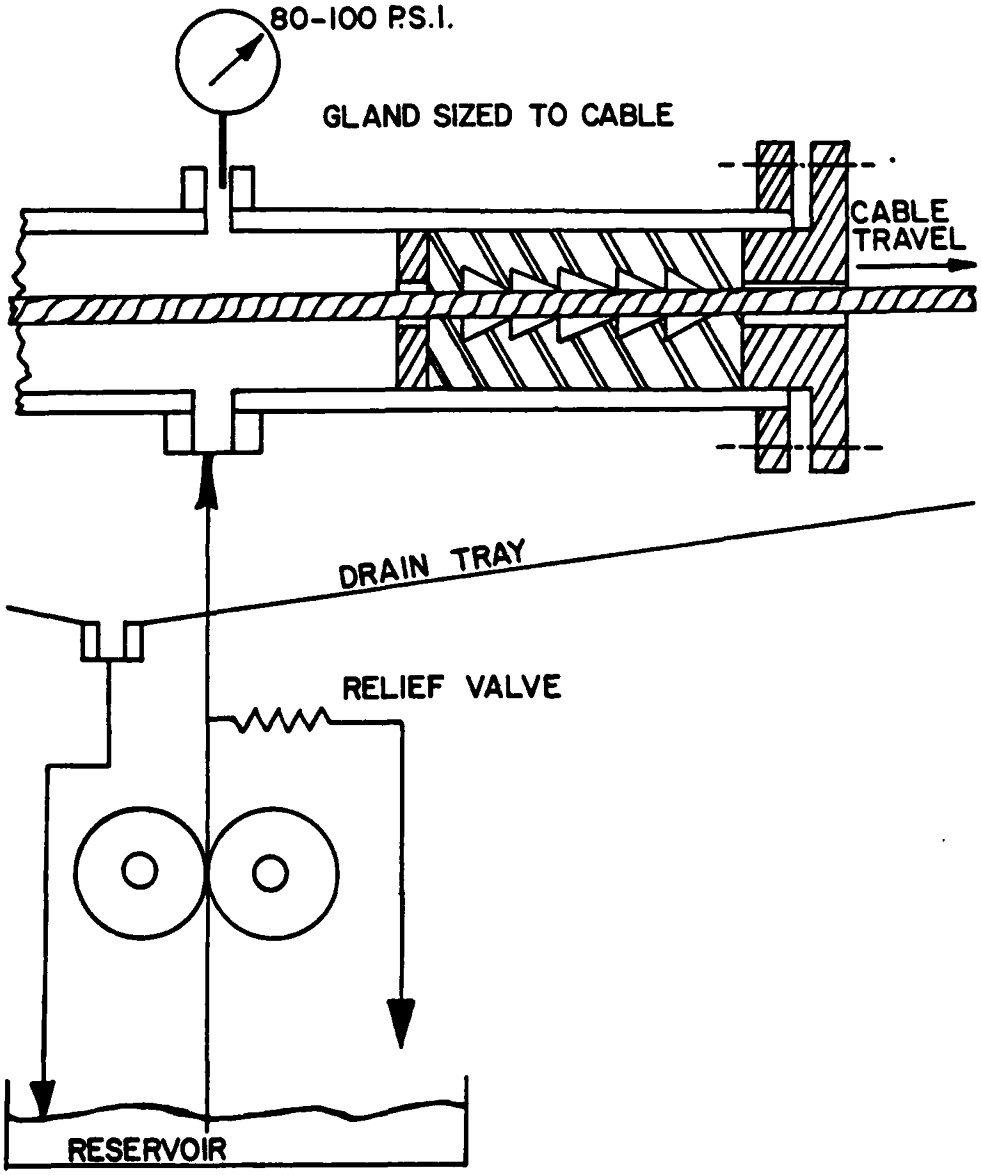
This marking means can be applied to any material having a high magnetic permeability and their detection life is known to be over a year.

FIGURE 2-35 REEL TURNER



VIEWED FROM ABOVE A CCW ROTATION
WILL TIGHTEN A LHL OUTER ARMOR

FIGURE 2-36 CABLE LUBRICATOR

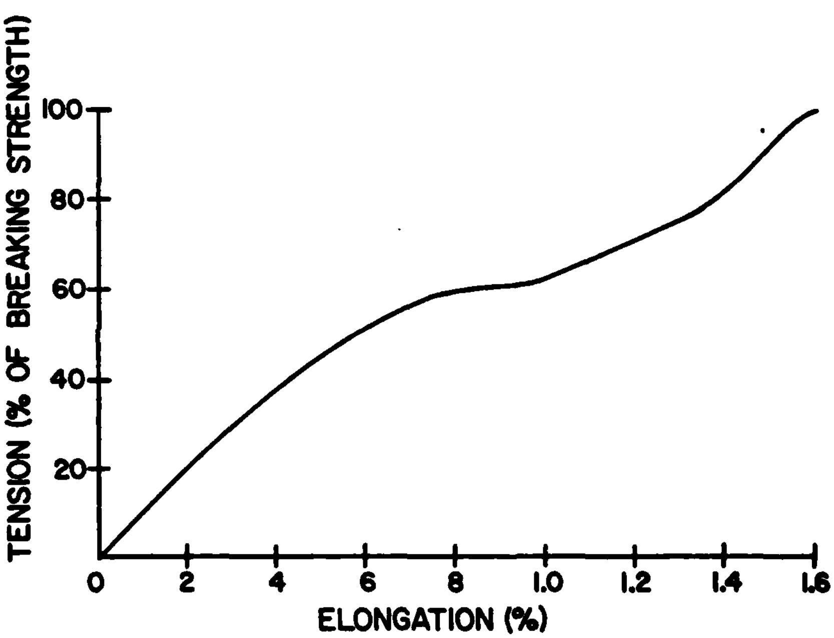


APPENDICIES

Number	Title
1.0	Typical Tension/Elongation Characteristic of a Double Armored Cable
2.0	Torque Ratio Equation
3.0	Properties of Stranded Copper Conductors
4.0	Copper Electrical Resistance Temperature Correction Multi-plier
5.0	Determining the Length of a Cabled Condutor
6.0	Sheave-to-Cable Bearing Pressure
7.0	Sheave-to-Cable Bearing Pressures of Typical Oceanographic Cables.
8.0	Properties of Insulating and Jacketing Materials
9.0	Properties of Improved and Extra Improved Plow Steel
10.0	Properties of Corrosion Resistant Armoring Materials
11.0	Elements of a Construction Specification
12.0	Contrahelically Armored Cable Specification Elements
13.0	Rereeling Setup
14.0	Cable Length Determination by the Conductor Resistance Method
15.0	Cable Length Determination by Weight
16.0	Derivation of Equation for Armor Layer and Net Armor Un-balanced Torque
17.0	Elongation of E-M Cables
18.0	Calculations for Physical Properties of E-M Cables
19.0	Principals of E-M Cable Splicing.
20.0	Armored Cable Diameter vs Breaking STrength.
21.0	Location of Short to Armor in Multiconductor Cables.

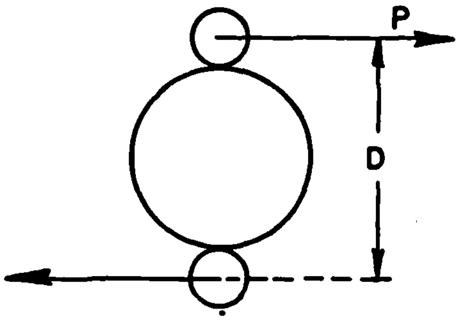
APPENDIX I.O

TYPICAL TENSION/ELONGATION CHARACTERISTIC OF A DOUBLE ARMORED CABLE

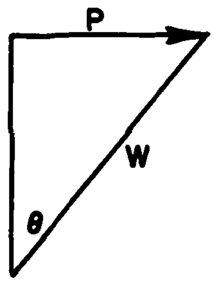


APPENDIX 2.0

TORQUE RATIO EQUATION

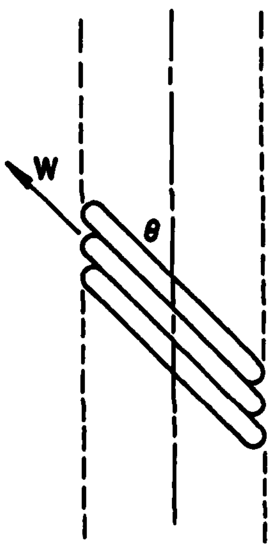


Torque = T = PD (1)



$$\sin\theta = \frac{P}{W}$$

$$W = \frac{P}{\sin\theta} \qquad (2)$$



For a wire, Youngs Modulus is:

$$E = \frac{\frac{W}{A}}{\varepsilon} \qquad W = EA\varepsilon \qquad (3)$$

using (2)

$$\frac{P}{\sin\theta} = EA\varepsilon \qquad (4)$$

$$P = EA\varepsilon \sin\theta \qquad (5)$$

substitute (5) in (1)

$$T = EA\varepsilon D \sin\theta \qquad (6)$$

summing the torque for all wires in an armor layer

$$\Sigma T = NEA\varepsilon D \sin\theta \qquad (7)$$

Express the ratio of the torques of outer and inner armor layers

$$R_T = \frac{\Sigma T_O}{\Sigma T_I} = \frac{N_O\ E_O\ A_O\ \varepsilon_O\ D_O \sin\theta_O}{N_I\ E_I\ A_I\ \varepsilon_I\ D_I \sin\theta_I} \tag{8}$$

E and ε can be considered the same for the inner and outer armors.

$$\therefore\quad R_T = \frac{N_O\ A_O\ D_O \sin\theta_O}{N_I\ A_I\ D_I \sin\theta_I} \tag{9}$$

$$A = \frac{\pi}{4} d^2 \tag{10}$$

substitute (10) in (0) and cancel the constant

$$\therefore\quad R_T = \frac{N_O d_O^{\,2}\ D_O \sin\theta_O}{N_O d_O^{\,2}\ D_O \sin\theta_O} \tag{11}$$

Note that this analysis is based on "P," the tangential force of armor wires, not onc able tension.

A = Cross-sectional area of an armor wire	(in^2)
d = armor wire diameter	(in)
D - pitch diameter of an armor layer	(in)
E = Young's Modulus	($\frac{lb}{in^2}$)
N = number of armor wires per layer	
P = tangential force of armor wires	(lb)
R_T = Torque ratio	
T = torque	(lb-in)
w. = tension in a wire	(lbf)
θ = armor lay angle	(degrees)
ε = armor wire strain	($\frac{in}{in}$)

APPENDIX 3.0

PROPERTIES OF STRANDED COPPER CONDUCTORS

Conductor Size	Stranding		Overall Diameter		Copper Area		Weight		Strand Break Strength		DC Resistance at 20^{o}C (68^{o}F)	
Awg	Inch	mm	Inch	mm	Cir,mils	Sq. mm	lbs/mft	Kg/Km	lbs	N	ohms/mft	ohms/Km
24	7/.008	7/.203	.024	.610	448	.227	1.38	2.06	12.7	56.3	24.0	78.6
	19/.005	10/.127	.024	.617	475	.241	1.47	2.18	13.4	59.7	22.6	74.1
23	7/.009	7/.229	.027	.686	567	.287	1.75	2.60	16.0	71.3	18.9	62.1
22	7/.010	7/.254	.030	.762	700	.355	2.16	3.21	19.8	88.0	15.3	50.3
	19/.0063	19/.160	.031	.778	754	.382	2.33	3.46	21.3	94.8	14.2	46.7
20	7/.0126	7/.320	.038	.960	1111	.563	3.43	5.10	31.4	139.8	9.7	31.7
	19/.0071	19/.180	.035	.876	958	.485	2.96	4.40	27.1	120.5	11.2	36.8
	19/.008	19/.203	.039	.988	1216	.616	3.75	5.58	34.4	152.9	8.8	29.0
19	7/.0142	7/.361	.043	1.082	1411	.715	4.36	6.48	39.9	177.5	7.6	24.9
	19/.0089	19/.226	.043	1.099	1505	.763	4.65	6.91	42.6	189.3	7.1	23.4
18	7/.0152	7/.386	.046	1.158	1617	.819	4.99	8.73	45.7	203.4	6.6	21.8
	19/.010	19/.254	.049	1.234	1900	.963	5.86	89.73	53.7	239.0	5.6	18.5
	41/.0063	41/.160	.051	1.295	1627	.825	5.02	7.47	46.0	204.7	6.6	21.6
16	7/.020	7/.508	.060	1.524	2800	1.42	8.64	12.86	79.2	352.1	3.8	12.6
	19/.0112	17/.284	.054	1.383	2382	1.21	7.36	10.95	67.4	299.7	4.5	14.8
14	7/.0242	7/.615	.073	1.844	4099	2.08	12.65	18.83	116	516	2.6	8.6
	19/.0142	19/.361	.069	1.753	3831	1.94	11.82	17.59	108	482	2.8	9.2

APPENDIX 3.0 (Cont'd)

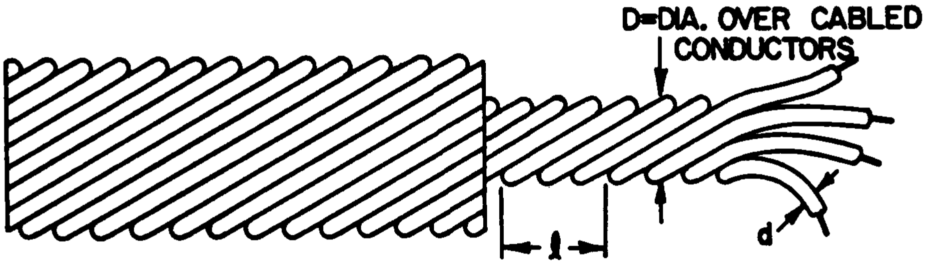
Conductor Size	Stranding		Overall Diameter		Copper Area		Weight		Strand Break Strength		DC Resistance at 20°C (68°F)	
Awg	Inch	mm	Inch	mm	Cir,mils	Sq. mm	lbs/mft	Kg/Km	lbs	N	ohms/mft	ohms/Km
12	7/.0305	7/.775	.092	2.324	5612	3.08	20.10	29.91	184	819	1.6	5.4
	19/.0179	19/.455	.087	2.210	6088	3.08	18.79	28.0	172	766	1.8	5.8
	19/.0185	19/.470	.090	2.284	6503	3.29	20.07	29.9	184	818	1.65	5.42
10	7/.0385	7/.978	.116	2.934	10376	5.26	32.02	47.65	293	1305	1.03	3.39
	19/.0234	19/.594	.114	2.889	10403	5.27	32.11	47.78	294	1308	1.03	3.39
9	7/.0432	7/1.097	.130	3.292	13064	6.62	40.32	50.00	369	1642	.82	2.70
	19/.0242	19/.615	.118	2.987	11127	5.64	34.34	51.10	3155	1499	.96	3.76
8	7/.0486	7/1.234	.146	3.703	16534	8.38	51.03	75.93	315	1399.	.65	2.13
	19/.0295	19/.749	.143	3.642	16535	8.38	51.03	75.94	468	2079.	.65	2.13
6	7/.0612	7/1.554	.184	4.663	26218	13.29	80.92	120.4	741	3297.	.41	1.34
	19/.0372	19/.945	.181	4.592	26292	13.32	81.15	120.8	743	3307	.41	1.34
4	7/.0772	7/1.961	.232	5.883	41719	21.14	128.8	191.6	1180.	5247.	.26	.84
	61/.0253	61/.643	.228	5.788	39045	19.78	120.5	179.3	1104.	4911.	.27	.90
3	61/.0285	61/.724	.257	5.784	49547	25.11	152.9	227.6	1401.	6231.	.22	.71

NOTE: All values nominal. 19 strand copper diameters are bunched construction. Break strength is based on approximately 40000 PSI for soft annealed copper. DC resistance of conductors in cable may vary due to helix or slight elongation.

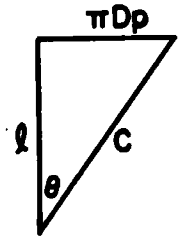
APPENDIX 4.0

COPPER ELECTRICAL RESISTANCE TEMPERATURE CORRECTION MULTIPLIER

Measurement Copper Temp. °C	°F	Mulitiplier
0	32	1.084
5	41	1.061
10	50	1.040
15	59	1.020
20	68	1.000
25	77	0.981
30	86	0.963
35	95	0.945
40	104	0.928
45	113	0.912
50	122	0.896
55	131	0.881
60	140	0.866
65	149	0.852
70	158	0.838
75	167	0.825
80	176	0.812



1. MEASURE THE CABLED CORE DIA. (D), CONDUCTOR DIAMETER (d) AND LAY LENGTH (ℓ)



LAY DIAGRAM I

2. USING THE MEASURED "Dp" AND "ℓ" CALCULATE THE LAY ANGLE (θ) USING THE LAY DIAGRAM

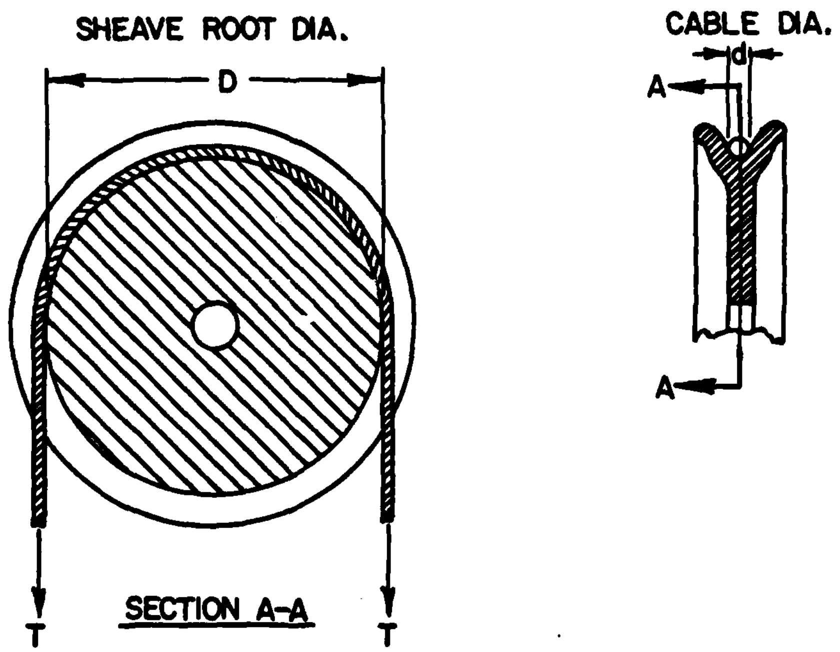
$$\theta = \arctan \frac{\pi D p}{\ell}$$

3. REFERRING TO THE LAY DIAGRAM DETERMINE THE RELATIONSHIP BETWEEN CABLE LENGTH "ℓ" AND CABLED CONDUCTOR "C"

$$L = C \cos\theta = C \cos\left[\arctan\left(\frac{\pi D p}{\ell}\right)\right]$$

APPENDIX 5.0 DETERMINING THE LENGTH OF A CABLED CONDUCTOR

APPENDIX 6.0 SHEAVE TO CABLE BEARING PRESSURE



PRESSURE $(p) = \frac{1}{2} T \times \frac{1}{A} = \frac{T}{2A}$

A = PROJECTED AREA OF CABLE CONTACT SURFACE = pD

$\therefore\ p = \frac{T}{2pD} \left(\frac{lbf}{in^2}\right)$

FOR WIRE ROPE USE WITH CAST CARBON STEEL SHEAVES THIS BEARING PRESSURE CAN ACCEPTABLY BE AS HIGH AS 1800 LBF/sq.in.

APPENDIX 7.0

SHEAVE-TO-CABLE BEARING PRSSURES OF TYPICAL OCEANOGRAPHIC CABLES

(d) O.D. (In)	(D) Sheave root (In)	(T) 20% of UTS (LBF)	(P) Bearing Pressure (LBF/Sq.In)
.183	10	580	158
.305	13	1,480	275
.224	12	880	164
.254	14	1,100	155
.282	16	1,440	160
.351	19	2,080	148
.375	20	2,360	157
.421	23	3,200	165
.670	28	6,600	164
.726	30	8,000	183

Calculated from the formula $P = \frac{T}{2pD}$
from Appencix 3.0

APPENDIX 8.0

Properties of Insulating and Jacketing Materials

	Polypropylene (PP)	Nylon 610 (N)	Polyethylene High Density (HDPE)	Polyethylene Low Density (LDPE)	FEP Tefton (TF)	Polyurethane (PU)	Hypalon (HY)	Thermoplastic Rubber (TPR)
Specific Gravity	.902	1.08	.947	.920	2.16	1.25	1.	0.88
Ultimate Tensile Strength, psi	5,000	8,000	3,400	2,200	3,000	6,000		
Ultimate Elongation, %	200	200	250	625	250	600		
Dielectric Constant, 1 KHz	2.22	4.5	2.32	2.25	2.1	7.5		
Rated Minimum Temperature, °C	-10	-40	-65	-65	-65	-55		
Relative Cost	0.4	1.2	0.4	0.4	12.0	1.8		

APPENDIX 9.0

PROPERTIES OF IMPROVED AND EXTRA IMPROVED PLOW STEEL

	(IPS) IMPROVED PLOW STEEL BREAKING STRENGTH (LB)		(IPS) IMPROVED PLOW STEEL TENSILE STRENGTH (PSI)		(XIPS) EXTRA IMPROVED PLOW STEEL BREAKING STRENGTH (LB)		(XIPS) EXTRA IMPROVED PLOW STEEL TENSILE STRENGTH (PSI)	
DIA	MIN	MAX	MIN	MAX	MIN	MAX	MIN	MAX
.012	29.1	33.5	257	296	32.1	38.5	284	341
.014	39.4	45.4	256	295	43.5	52.2	283	338
.017	57.9	66.7	255	294	63.8	76.7	281	
.018	64.9	74.5	255	293	71.4	85.8	281	337
.022	96.2	111	253	291	106	127	280	335
.0245	119	137	252	291	132	158	280	335
.027	144	166	252	290	159	191	278	333
.028	155	178	251	289	171	205	278	333
.030	177	204	250	289	195	235	276	333
.032	201	232	250	289	222	266	276	331
.033	214	246	250	288	236	283	276	331
.035	241	276	251	287	265	318	276	331
.037	268	308	249	287	295	354	274	329
.0385	290	332	249	285	319	383	274	329
.040	312	359	248	286	344	413	374	329
.042	344	395	248	285	378	454	273	328
.0435	368	424	248	285	406	487	273	328
.047	424	492	244	284	472	567	272	327
.048	447	514	247	284	492	591	272	327
.049	466	535	247	284	512	615	272	326
.050	483	555	246	283	533	640	272	326
.051	503	578	246	283	554	665	271	326
.054	563	646	246	282	619	744	270	325
.056	603	695	245	282	665	799	270	324
.0575	636	732	245	282	700	841	270	324
.059	670	768	245	281	736	885	269	324
.065	806	929	243	280	890	1070	268	323
.068	882	1020	243	281	972	1170	268	322
.070	935	1070	243	278	1030	1240	268	322
.074	1040	1200	242	279	1150	1380	268	321
.077	1120	1290	241	277	1240	1490	266	320
.104	2010	2320	237	273	2220	3660	261	313

APPENDIX 10.0 PROPERTIES OF CORROSION RESISTANT ARMORING MATERIALS

	GIPS	NITRONIC 50	MP35N	ALLEGHENY 6X	INCONEL 625
density	7.8 gm/cc .281 lb/in^3	7.88 gm/cc .285 lb/in^3	8.57 gm/cc .309 lb/in^3	8.0 gm/cc .293 lb/in^3	8.44 gm/cc 0.305 lb/in^3
elec.resistivity at 70°F (21 C)	19 microhm-cm	82 microhm-cm	101 microhm-cm	82 microhm-cm	129 microhm-cm
UTS 60% cold reduction	270 Ksi 1862 Mpa	246 Ksi 1696 Mpa	280 Ksi 1793 Mpa	205 Ksi 1412 Mpa	269 Ksi 1853 Mpa
.2% offset yield	216 Ksi 1489 Mpa	234 Ksi 1613 Mpa	240 Ksi 1655 Mpa	184 Ksi 1270 Mpa	253 Ksi 1743 Mpa
Modulus of elasticity	30×10^6 psi 206,844 Mpa	28×10^6 psi 193,054 Mpa	33.6×10^6 psi 193,100 Mpa	29×10^6 200,000 Mpa	30×10^6 psi 206,844 Mpa
Coefficient of thermal expansion 0-400 F (0-204 C)	.0063 ft/M'/F° .0113 m/km/c°	.0092 ft/M'/F° .0162 m/Km/c°	.0076 ft/M'F° .0137 m/Km/c°	.0089 ft/M'F° .0160 m/Km/c°	.0073 ft/M'/F° .0131 m/Km/c°
magnetic properties	magnetic	non-magnetic	non-magnetic	non-magnetic	non-magnetic
Analysis (Typical)					
Ni		12.5%			
Co			35%	24.50%	57.0 Ni
Cr			35%	-	1.0 Co
Mo			20%	20.25%	21.0 Cr
Mn		2.2	10%	6.25%	9.0 Mo
Material Cost Ratio*	1	5.0	.	1.50%	0.5 Mn
Cable Cost Ratio *	1	8	65	19	41
		3	16	6.5	11

The above data are presented for comparison only and are not intended as accurate design or quotation data.

Albert G. Berian
10 July 1980

APPENDIX 11.0

ELEMENTS OF A CONSTRUCTION SPECIFICATION

Electrical

Conductor size and stranding insulation and thickness

Jacket thickness, if used

Insulation resistance

Capacitance

Dielectric strength

Mechanical

Cable length and tolerance

Number of wires and diameter

Breaking strength, minimum

Permissable number of welds in the finished armor wire

Weight of zinc coating on wires

Overall diameter and tolerance

Test Data to be furnished

Conductor resistance

Capacitance

Dielectric strength

Insulation

Insulation resistance

APPENDIX 12.0

CONTRA-HELICALLY ARMORED CABLE SPECIFICATION ELEMENTS

1.0 SCOPE

1.1 System types include:

- tow
- umbilical
- vertical array
- floating
- bottom deployed.

1.2 Mission profiles cover:

- speed of deployment
- frequency and duration of deployment
- steady-state and varying tensions
- known hazards such as fishbite, corrosive conditions and mechanical impact on the cable such as crushing, abrasion, etc.
- environmental
- geographical location

1.3 Type of handling system

- which type (storage or tension)
- winch characteristics (drum dia., traverse, fleet angle, drum grooving, level wind)
- sheave dia. and groove design (geometry, smoothness, hardness, coating used)
- coiling into bales, cannisters, tanks, etc.

2.0 REFERENCED DOCUMENTS

These could include:

- U.S. Government

MIL-C-915 - Shipboard cable frequently referenced for the non-hosing test requirement and conductor color coding.

MIL-W-16878 - Insulated Wire Covers

MIL-C-17; RF Cables

a basic document for RG cables

MIL-I-45208

covers basic quality assurance requirements

covers comprehensive quality assurance requirements.

MIL-C-24215

Underwater Electrical Connectors

Independent Power Cable Engineers Assn (IPCEA)

various sections cover insulation construction and other requirements for power cables to ____ KV.

3.0 REQUIREMENTS

3.1 Electrical

Power Conductors: phases, frequency, watts or current, max. voltage drop or resistance, corona initiation and suppression voltages.

Control: voltage, current

Signal or communication: characteristic impedance, resistance, voltage, crosstalk, attenuation capacitance.

3.2 Mechanical

- Tension: maximum steady-state working, maximum varyng tension, factor of safety.
- Diameter and tolerance
- Weight objective

  positive buoyancy
  neutral buoyancy
  maximum negative buoyancy
- Torque characteristics

  rotation per 1000 ft.
  torque ratio
- Bending fatigue

  number of cycles under specified conditions of d/D ratio, excursion speed

3.0 REQUIREMENTS (continued)

3.3 Cable Accessories (Supplied by cable manufacturer or purchaser)

- Electrical connector: Configuration, cable connector, type of seal to cable, bulkhead feed through.
- Mechanical Connector: Configuration, type of mechanical bending strain relief.
- Fairings: Length to be faired, description of fairing.

4.0 TEST

4.1 Qualification: may include evaluations of the high voltage and transmission properties of conductors, steady-state and under mechanical test conditions.

Mechanical testing may include:

- ultimate breaking strength
- flexure cycling
- torque balance
- vibration
- non-hosing hydrostatic
- hydrostatic pressure cycling
- elongation and diameter deformation

4.2 Acceptance

Electrical: IR, dielectric strength

Mechanical:

OD
Surface quality

5.0 MARKING AND SHIPPING

- method of marking cable

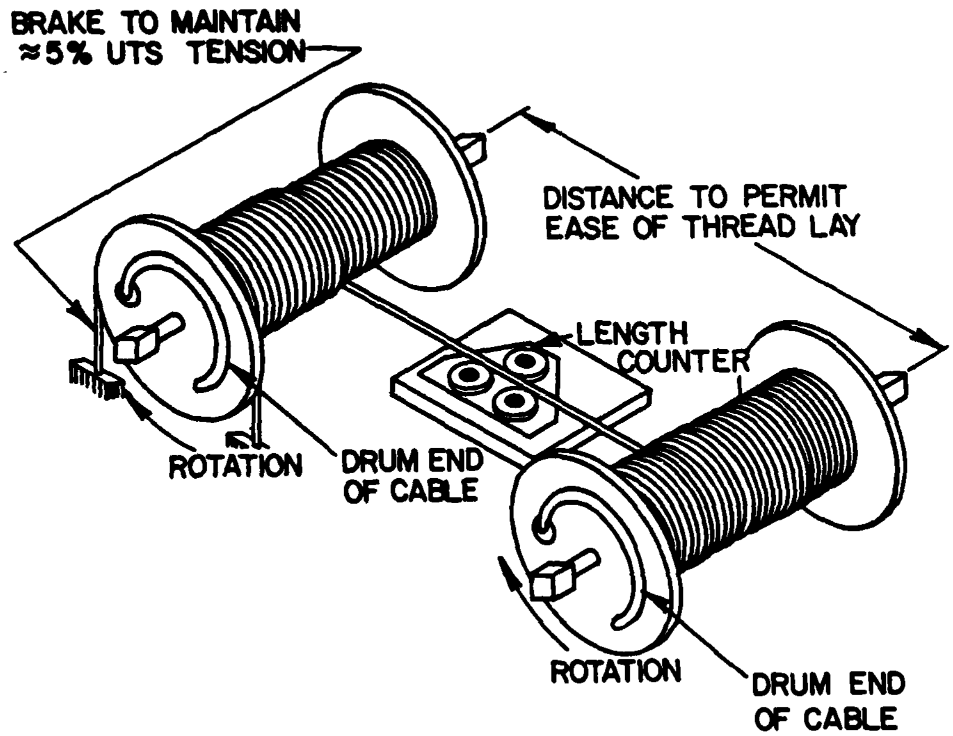
- reel requirements

  special lagging requirements

  marking of reels

- length of inner end of cable to be free

APPENDIX 13.0
REREELING SETUP



APPENDIX 14.0

CABLE LENGTH DETERMINATION BY CONDUCTOR RESISTANCE MEASUREMENT

A. CABLE HAVING A CENTER CONDUCTOR

1. Refer to the as-received conductor resistance and length.

2. Using the most accurate ohmmeter or resistance bridge available measure the center conductor resistance

3. The cable length is

where: L = present length

$$L = \frac{R}{R_m} = 1000 \text{ ft}$$

R - conductor resistance, ohms/1000 ft.

R_m = measured resistance

B. CABLE HAVING NO CENTER CONDUCTOR

1. Determine the relationship between the cabled conductor length and cable length using the procedure of Appendix 5.

2. As for a center conductor cable, measure the conductor resistance; this measurement will be of the cable conductor, C.

3. Calculate the present length:

$$L = C \cos \theta \frac{R}{R_m}$$

APPENDIX 15.0

CABLE LENGTH DETERMINATION BY WEIGHT

Although less accurate than the rereeling or resistance measurement methods it may be used when the approximate present length measurement is needed.

1. Determine the tare weight , W_r, of the reel.

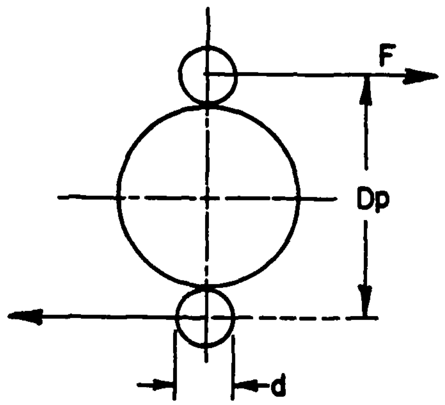
   Determine the weight per 1000 ft of cable, w, from manufacturer's data.

   Weight the reel of cable, W_2, and calculate the length by:

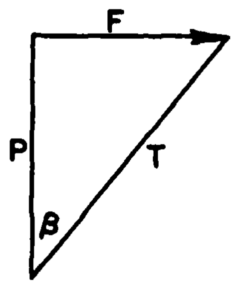
$$L = \frac{W_2 - W_r}{w} \times 1000 \text{ ft}$$

APPENDIX 16.0

E-M Cable Torque



$$\text{Torque} = M = F\ Dp \qquad (1)$$



$$F = P \tan \beta \qquad (2)$$
$$T = P \cos \beta$$

combining (1) and (2)

$$\text{Torque} = M = P\ Dp \tan \beta \qquad (3)$$

Hook's law for wires

$$E = \frac{\frac{T}{A}}{\varepsilon} \qquad E\varepsilon = \frac{T}{A}$$

assuming $E\varepsilon$ equal for the inner and outer armor layers

$$\frac{T_O}{A_O} = \frac{T_I}{A_I} \qquad (4)$$

express T in terms of P from (2)

$$\frac{P_O}{A_O \cos \beta_O} = \frac{P_I}{A_I \cos \beta_I}$$

$$(5)$$

$$\frac{P_O}{P_I} = \frac{A_O \cos \beta_I}{A_I \cos \beta_I}$$

sum for all armor wires

$$\frac{\Sigma P_o}{\Sigma P_I} = \frac{N_o\ A_o \cos \beta_o}{N_I\ A_I \cos \beta_I} \tag{6}$$

$$\Sigma P_o = \Sigma P_I \frac{N_o\ A_o \cos \beta_o}{N_I\ A_I \cos \beta_I} \tag{7}$$

$$\Sigma P = \Sigma P_o + \Sigma P_I \tag{8}$$

substitute (7) in (8)

$$\Sigma P = \Sigma P_I + \Sigma P_I \frac{N_o\ A_o \cos \beta_o}{N_I\ A_I \cos \beta_I} \tag{9}$$

$A = \frac{\pi}{4} d^2$ and the constant $\frac{\pi}{4}$ cancels

$$\Sigma P = \Sigma P_I \left(1 + \frac{N_o\ d_o^2 \cos \beta_o}{N_I\ d_I^2 \cos \beta_I}\right) \tag{10}$$

$$\Sigma P = \Sigma P_I \left(\frac{N_I\ d_I^2 \cos \beta_I + N_o\ d_o^2 \cos \beta_o}{N_I\ d_I^2 \cos \beta_I}\right) \tag{11}$$

$$\frac{\Sigma P_I}{\Sigma P} = \frac{N_I\ d_I^2 \cos \beta_I}{N_I\ d_I^2 \cos \beta_I + N_o\ d_o^2 \cos \beta_o} \tag{12}$$

$$\Sigma P_I = \Sigma P \frac{N_I\ d_I^2 \cos \beta_I}{N_I\ d_I^2 \cos \beta_I + N_o\ d_o^2 \cos \beta_o} \tag{13}$$

$$\Sigma P_o = \Sigma P \frac{N_o\ d_o^2 \cos \beta_o}{N_o\ d_o^2 \cos \beta_o + N_I\ d_I^2 \cos \beta_I} \tag{14}$$

To obtain an expression for inner and outer armor torques, substitute (13) and (14) respectively into (3).

$$\Sigma M_I = \Sigma P \frac{N_I\ d_I^2 \cos \beta_I}{N_I\ d_I^2 \cos \beta_I + N_o\ d_o^2 \cos \beta_o} (D_I \tan \beta_I) \tag{15}$$

$$\Sigma M_o = \Sigma P \frac{N_o\, d_o^{\,2} \cos\beta_o}{N_o\, d_o^{\,2} \cos\beta_o + N_I\, d_I^{\,2} \cos\beta_I} (D_o \tan\beta_o) \quad (16)$$

$$\Sigma M = M_o - M_I \quad (17)$$

Substitute (15) and (16) into (17)

$$\Sigma M = \Sigma P \frac{N_o\, d_o^{\,2}\, D_o \cos\beta_o \tan\beta_o - N_I\, d_I^{\,2}\, D_I \cos\beta_I \tan\beta_I}{N_o\, d_o^{\,2} \cos\beta_o + N_I\, d_I^{\,2} \cos\beta_I} \quad (18)$$

$$\Sigma M\Big|_{d_o = d_I} = \Sigma P \frac{N_o\, D_o \cos\beta_o \tan\beta_o - N_I\, D_I \cos\beta_I \tan\beta_I}{N_o \cos\beta_o + N_I \cos\beta_I} \quad (19)$$

$$\Sigma M\Big|_{N_o = N_I} \Sigma P \frac{D_o\, d_o^{\,2} \cos\beta_o \tan\beta_o - D_I\, d_I^{\,2} \cos\beta_I \tan\beta_I}{d_o^{\,2} \cos\beta_o + d_I^{\,2} \cos\beta_I} \quad (20)$$

F = component of armor wire tension tangential to cable axis (lb/ft)

Dp = pitch diameter of wire (in)

T = armor wire tension (lb/ft)

P = component of armor wire tension axial to cable axis (lb/ft)

β = armor lay angle (degrees)

M = torque (lb/in)

ε = wire strain (in/in)

subscripts:

o = outer armor

I = inner armor

APPENDIX 17.0

LOAD vs ELONGATION
(at 50% of Breaking Strength)

Dia.	No. of Conductors	Armor	% Elongation at 50% BS
.185	1	12/.0355 12/.022	0.50%
.206	1	15/.033 9/.033	0.80%
.221	1	15/.0355 11/.031	0.48%
.319	1	18/.044 12/.044	0.66%
.428	1	18/.059 18/.042	1.10%
.376	7	23/.042 17/.042	0.60%
.427	7	18/.059 18/.042	0.70%
.464	7	24/.049 24/.039	0.64%
.520	7	20/.064 19/.052	0.61%

APPENDIX 18.0

Calculations for Physical Properties of E. M. Cables

Wa = weight in air (lb/M^1)

B = buoyancy (lb/M^1)

Pw = density of sea water (lb/ft^3) ~64 lb/ft^3

Ac = cable cross-section area (in^2)

d = cable dia. (in)

M^1 = 1,000 ft

Scw = specific gravity of cable in sea water

Pca = specific gravity

T_B = breaking strength

Ww = weight in water

1.0 Weight in sea water (Ww) of jacketed cables (armored or unarmored)

$$Ww = Wa - B$$

$$B = 1{,}000 \text{ ft} \times Ac \times Pw$$

$$B = 1{,}000 \times \frac{\frac{\pi}{4}d^2}{144} \times 64 \frac{16}{ft^3}$$

$$B = 349d^2$$

$$Ww\Big|_{\text{Jacketed}} = Wa - 349d^2$$

2.0 Weight in sea water of non-jacketed, armored cable approximately 10% void in armor interstices

$$Ww\Big|_{\text{non-jacketed}} = Wa - 314d^2$$

3.0 Specific gravity in sea water (jacketed)

$$Scw = \frac{Pc\ a}{Pw}$$

$$Pca = \frac{144\ Wa}{\frac{\pi}{4}\ d^2 \times 1{,}000}$$

$$Scw = \frac{144Wa}{\frac{\pi}{4} d^2 \times 1{,}000 \times 64} = \frac{0.2865Wa}{100d^2}$$

4.0 Specific gravity in sea water (non-jacketed)
(Approximately 10% voids in armor interstices)

$$Scw = \frac{144 \times Wa \times 0.9}{\frac{\pi}{4} d^2 \times 1{,}000 \times 64} = \frac{.2578Wa}{100d^2}$$

5.0 Strength-to-weight ratio in water

$$= \frac{T_B}{Ww} \quad \frac{lb}{\frac{lb}{M^1}} = M^1$$

APPENDIX 19.0

PRINCIPLES OF E-M CABLE SPLICING

The following presentation is intended to only present the principles which apply to E-M cable splicing, not for a working procedure.

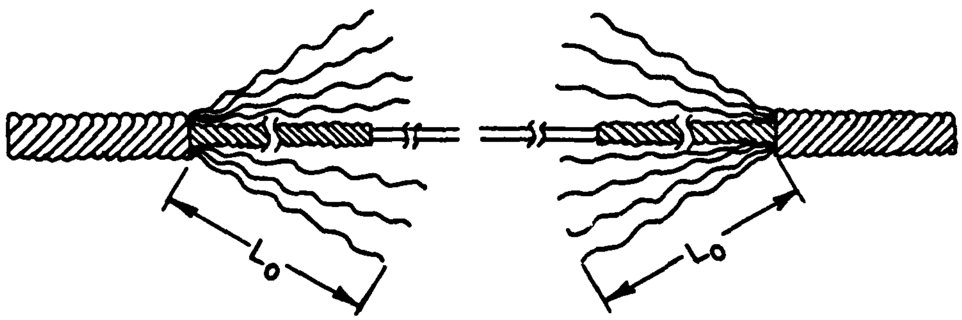
1. From the ends of the cable to be spliced unstrand the outer armor a distance from each end equal to:

$$L_O = 6N_O l_O$$

where L_O = Unstranded length of outer armor wires

l_O = lay length of outer armor

N_O = number of outer armor wires

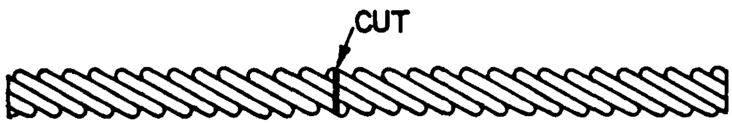


For convenience in handling the unstranded wires may be taped in groups of three.

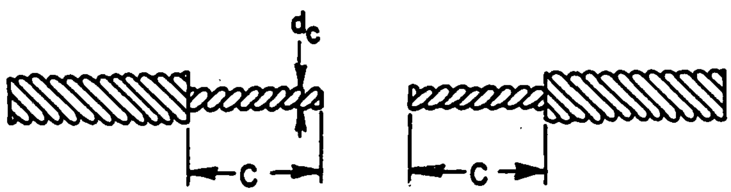
2. Using the above length determination unstrand the inner armor wires and tape them. The shorter lay length of the inner armor will result in $L_I < L_O$.

3. Note the rifling, or helical grooving, which the inner armor has compressed on the core. It is desirable to replace the inner armor wires into this grooving.

4. Square cut the core so that there is a continuity of the helical grooving.



5. Prepare the insulated conductors for splicing by cutting the insulation at a distance of approximately 20 times the conductor diameter shown in Appendix 3.0.



$c = 20\ d_C$

where C = length of stripped conductor

d_C = diameter of conductor

6. Prepare a conical surface on the ends of the insulation with the height of the truncated cone being:

$I = T\ d_I$

where I = Height of truncated cone

d_I = Insulation diameter

7. Separate the conductor wires in preparation for splicing. Two splicing methods are used:

   (a) combing and tying
   (b) soldering

   The soldering method consists of combing, or enmeshing the wires and, using a minimum amount of solder, bond the wires without allowing the solder to wick along the stranded conductor. A long soldered joint creates a stiff section having end discontinuities which form points of stress concentration.

   The combing and tying procedure for conductor splicing has two versions which are in popular use:

   (a) comb complete strand to obtain a splice cross-section having twice the number of wires as the basic strand.

   (b) cut wires from each of the ends to be spliced so that the number of wires in the splice cross-section is the same as that of the basic strand.

   When the wires of each strand are combed, or intermeshed, the strands not just overlapping, tie them using the lowest denier, unwaxed dental floss available. This tying is intended to bind the wires to obtain a few pounds pull-out strength but allow a length adjustment of the spliced conductor when the completed cable splice is tensioned.

   The insulation splice void is filled with wraps of commercially-available, self-adhesive Teflon tape which is 0.0005 in. thick and 0.25 wide. Avoiding wrinkles or other discontinuities, the wrapping uses a 50% overlap of the tape. Continue wrapping to maintain an increasing uniform diameter. As the diameter increases the taped length will also in-

crease until the splice diameter reaches the original insulation diameter.

8. The inner armor wires are restranded into the original grooving in the core. The wire splices should alternate between ends; i.e., a short wire of one end should adjoin a long one of the same end. The wire splices should be separated by a distance of about five times the lay length.

   When the restranding operation is completed the wire splice locations can be reviewed prior to cutting the wires and laying them in place. It is not necessary to bond the inner armor wires.

9. The outer armor wires are restranded using the same proedure as for the inner armor. There are four procedures in use for treatment of the wire ends of the outer armor; they are:

   (a) butted only as is the inner armor.

   (b) soldered, whereby the butted wires are silver soldered to the adjoining wires.

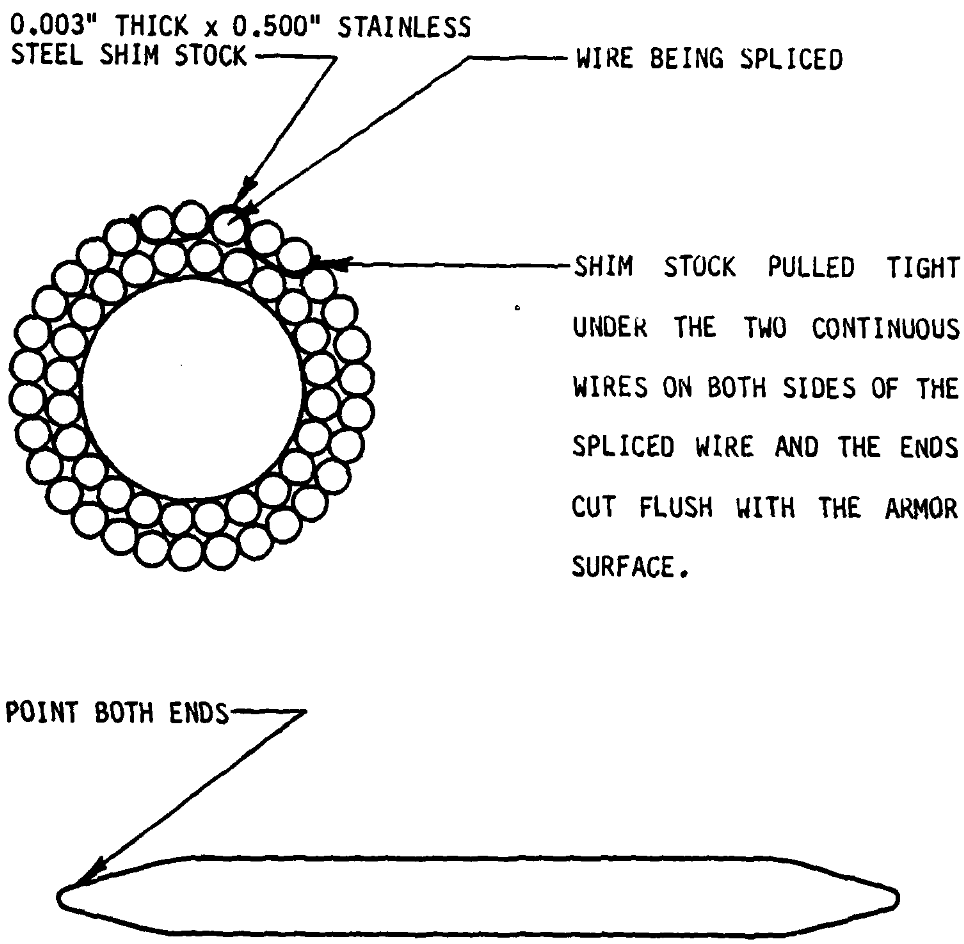
   (c) shim stock spliced as illustrated on the page.

   (d) butt welded. This procedure is used to join broken wires during the armoring process as discusses in Par. 5.8 but is rarely sed as a part of cable repair procedure.

10. The diameter of the spliced section will usually be found to be larger than an unspliced section. It is therefore desirable to condition the splice before releasing the cable for service. This can be accomplished by running the cable over a sheave several times; somewhat duplicating the manufacturer's prestressing procedure as described in Par. 5.9. The cable diameter as well as electrical continuity should be monitored.

    Service shops may use offset rolls for this prestressing procedure; they are depicted in Fig.

SHIM STOCK SPLICE

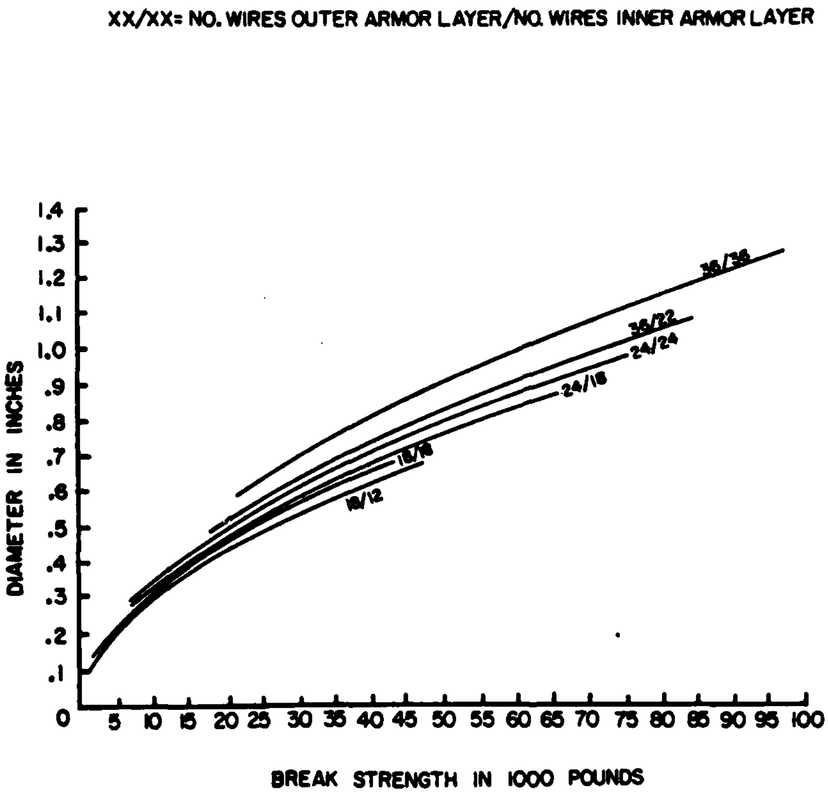


0.500" x 0.003" STN STL

SHIM STOCK

APPENDIX 20.0

ARMORED CABLE DIAMETER vs BREAKING STRENGTH

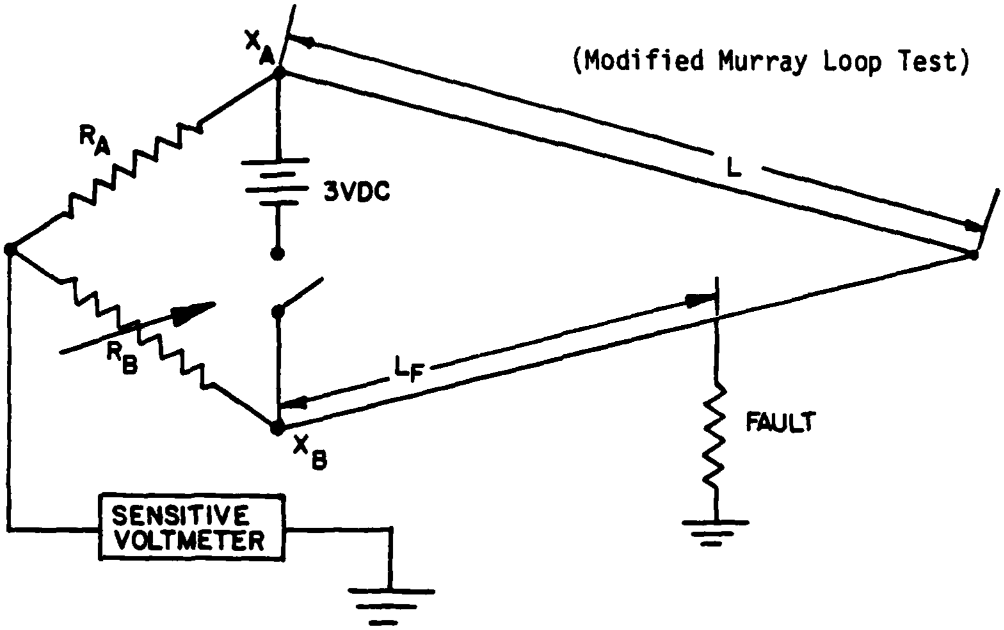


The above curves indicate Outer Diameter verses Break Strength for various double armor construction. The data is based on using standard galvanized improved plow steel tensiles. For example, a 3/4" diameter cable, with 24 outer wires and 18 inner wires, has approximately 46,000 lbs ultimate break strength.

APPENDIX 21.0

LOCATION OF SHORT TO ARMOR IN MULTICONDUCTOR CABLES

(Modified Murray Loop Test)



APPENDIX 21.0
LOCATION OF SHORT TO ARMOR
IN MULTICONDUCTOR CABLES.

where R_A α R_B = Resistance of bridge arms at balance

L = length of each conductor

L_F = distance to fault from X_B

R = resistance of an unfaulted conductor of length L

R_F= resistance of conductor length L_F from X_B to the fault

Ratio the two sides of the bridge

or

Express R_F and R in terms of length equivalents

BIBLIOGRAPHY

Handbook of Electric Cable Technology for Deep Ocean Applications.

Rotary Shaft-Seal Handbook for Pressure Equalized, Deep Ocean Equipment, NSRDC (A), 7-573, Oct. 71, NTIS, AD-889-330 (L).

Handbook of Vehicle Electrical Penetrators, Connectors, and Harnesses for Deep Ocean Applications, July 71, NTIS, AD-888-281.

Handbook of Fluids and Lubricants for Deep Ocean Applications, NSRDC (A), MATLAB 360, Revised 72, NTIS, AD-893-990.

Handbook of Fluid-Filled, Depth/Pressure Compensating Systems for Deep Ocean Applications, NSRDC (A), 27-8, April 72, NTIS, AD-894-795.

Handbook of Electrical and Electronic Circuit-Interrupting and Protective Devices for Deep Ocean Applications, NSRDC (A), 6-167, Nov. 71, NTIS, AD-889-929.

Handbook of Underwater Imaging Systems Design, NUCTP 303, Jul. 72, NTIS, AD-904-472 (L).

Handbook of Pressure-Proof Electrical Harness and Termination Technology for Deep Ocean Applications, Oct. 74, NTIS No. not assigned.

Cable Design Guidelines Based on a Bending, Tension and Torsion Study of an Electromechanical Cable, NUSC Technical Report, 4619, Rolf G. Kasper, Engineering Mechanics Staff.

Handbook of Electric Cable Technology for Deep Ocean Applications, NSRDL(A), 6-54/70, November 1970. AD 877-774.

Rotary Shaft-Seal Selection Handbook for Pressure Equalized, Deep Ocean Equipment, NSRDC(A), 7-573, October 1971. AD 889-330(L).

Handbook of Verhicle Electric Penetrators, Connectors and Harness for Deep Ocean Applications, July 1971. AD 888-281.

Hanbook of Fluids and Lubricants for Deep Ocean Applications, NSRDC(A), MATLAB 360, Revised 1972. AD 893-990.

Handbook of Fluid Filled, Depth-Pressure Compensating Systems for Deep Ocean Applications, NSRDC(A), 27-8, April 1972. AD 894-795.

Handbook of Electrical and Electronic Circuit-Interrupting and Protective Devices for Deep Ocean Applications, NSRDC(A), 6-167, November 1971. AD 889-929.

Handbook of Underwater Imaging Systems Design, NUTP 303, July 1972. AD 904-472(L).

Capadona, Emanuel A. Preformed Line Products Co., Cleveland, Ohio 406130, Dynamic Testing of Load Handling Wire Rope and Synthetic Rope. 14 Feb 69 - 15 Jan 70 - NTIS, AD-712-486. 3.00, 59 p.

Vanderveldt, Hendrikus H., DeYoung, Ron. Catholic University of America, Washington, D.C., Institute of Ocean Science α Engineering 404847, A Survey of Publications on Mechanical Wire Rope and Wire Rope Systems, NTIS, AD-710-806. 3.00, Aug 70.

Vanderveldt, Hendrikus H., Laura, Patricia A., Gaffney, Paul G., II. Catholic University of America, Mechanical Behavior of Stranded Wire Rope. July 69, NTIS, AD-710-805. 3.00, 59p.

Powell, Robert B. All American Engineerig Co., Wilmington, Delaware 01800, A Study of the Causes of Wire Rope and Cable Failure in Oceanographic Service; Sept 67, NTIS, AD-658-871, 6.00, 43p.

Czul, E. C., Germani, J. J. NRL, Washington, D.C., Load-Carrying Terminals for Armored Electric Cables. NTS, AD-621-564, 3.00, 31 Aug 65.

Milburn, D. A., Rendler, N. J. Methods of Measuring Mechanical Behavior of a Wire Rope, NRL, Washingotn, D.C., 25195. NTIS, AD-745-737, June 72, 3.00, 29p.

Heller, S. R., Jr., Matanzo, Frank, Metcalf, John T. Catholic University of America, Washington, D. C. 406291, Axial Fatigue of Wire Rope in Sea Water. NTIS, AD-743-924, 15 June 72, 3.00, 75p.

Case, R. O. Alabama University, Bureau of Engineering Research 067500, Research Program to Determine Fatigue Properties of Wire Rope Having Individually Coated Wires. NTIS, AD-740-591 30 Oct 66, 3.00, 15p.

Gambrell, S. C., Jr. Alabama University, Bureau of Engineering Research 067500, Effects of Various Connectors on Fatigue Life of Wire Rope. NTIS, AD-740-389, 10 April 69, 37p.

Black, Robert. Naval Air Engineering Center, Philadelphia, PA 403208, MK 7 Arresting Gear Purchase Cable Development Program, July 1969 through Dec 1970. NTIS, AD-733-988, 24 Nov 71, 3.00.

Heller, S. R., Jr., Matanzo, F. Catholic University of America Washington, D. C. 406291, Axial Fatigue of Wire Rope. 25 June 71. NTIS, AD-726-457, 3.00, 43p.

Casarello, M. J. Catholic University of America, Washington, D.C. 404347, Institute of Ocean Science and Engineering, Workshop on Marine Wire Rope Held at Catholic University of America, Washington, D.C. 11-13 Aug 70. NTIS, AD-791-373, 3.00, 106p.

Gibson, Phillip T., Larson, Charles H, Cross, Hobart A. NTIS AD-776-993/8 Battelle Columbus Labs, Long Beach, CA 40689, Determination of the Effect of Various Parameters on Wear and Fatigue of Wire Rope Used in Navy Rigging. 15 March 72, 8.50, 106p.

Durelli, A. J., Machida, S., Parks, V. J. Strains and Displacements on a Steel Wire Strand. Catholic University of America, Washington, D. C., Dept. Civil and Mechanical Engineering 406291. HTIS, AD-772-346/3, Reprint 1972, Naval Engineering Journal, Dec 72.

Milburn, Darrell A. Study of a Titanium Wire Rope Developed for Marine Applications Study. Naval Research Lab, Washington, D.C. 251950, 2 Nov 73, NTIS, AD-771-355/5, 2.75, 24p.

Minro, John C, Gibson, Phillip T., Cross, Hobart A. Helicopter Load Tension-Member Study. Battello Columbus Labs, Long Beach, CA 407629, HTIS, AD-755-532, 26 Jun 70 - 12 Apr 72, 3.00, 170 p.

Nowatzki, J. A. "Strength Member Design for Underwater Cables," 1971.

Schaner, D. S. "Ocean Applications of Wire Line Tension Measuring Devices," 1966.

Poffenberger, J. C. "Dynamic Testing of Cables," 1966.

Brainard, E. C., II. "Braiding Techniques Applied to Oceanographic Cables," 1967.

Haas, F. J. "Natural and Synthetic Cordage in the Field of Oceanography," 1967.

Capadona, E. A. "Establishing Test Parameters for Evaluation and Design of Cable and Fittings for High Speed Towed Systems," 1967.

Louzader, J. C. and Bridges, R. M. "Integration as Applied to Undersea Cable Systems," 1974.

Gibson, P. T. et al. "Evaluation of Kevlar-Strengthened Electromechanical Cable," 1974.

Nowak, G. and Bowers, W. E. "Computer Design of Electromechanical Cables for Ocean Applications," 1974.

Bridges, R. M. "An Airborne Sonar Cable - Design Problems and Their Solutions," 1974.

Briggs, E. "Electrical Distribution System for a Subsea Oil Producing and Pumping Station," 1974.

O'Brien, D. G. "An Update on Recommended Techniques for Terminating Underwater Electrical Connectors to Cables," 1974.

Berian, A. G. "An Impregnated High-Strength Organic Fiber for Oceanographic Strength Members," 1974.

Papers Presented by the Undersea Cable and Connector Committee of the Marine Technology Society 1966 to 1974

O'Brien, D. G. "Application of Glass-Hermetic-Sealed Watertight Electrical Connectors," 1967.

Walsh, D. K. "Underwater Electrical Cables and Connectors Engineered as a Single Requirement," 1966.

Capadona, E. A. et al. "Dynamic Testing of Cables," 1966.

Glowacz, A. and Louzader, J. "Thru Hull Electrical Penetrators," 1970.

Briggs, E. M. "The Design of a 4160 Volt Deep Sea Wet and Dry Conector System," 1970.

Hottel, H. C., Jr. "Expendable Wire Links," 1971.

Small, F. B. and Weaver, R. T. "Underwater Disconnectable Connector," 1971.

Tuttle, J. D. "Underwater Electrical Integrity," 1971.

Bridges, R. M. "Structural Requirements of Undersea Electrical Cable Terminations," 1971.

Edwards, F. L. and Patterson, R. A. "Pressure Balanced Electrical Hull Penetrators and External Cabling for Deepstar 20,000," 1970.

Saunders, W. "Pressure-Compensated Cable," 1972.

McCartney, J. F. and Wilson, J. V. "High Power Transmission Cables and Connectors for Undersea Vehicles," 1971.

Milburn, D. A. and Rendler, N. J. "Methods of Measuring the Mechanical Behavior of Wire Rope," 1972.

Noonan, B. J. and Casarella, M. J. and Choo, Y. "An Experimental Study of the Motion of a Towline Attached to a Vehicle Moving in a Circular Path," 1972.

Zamick, E. E. and Casarella, M. J. "The Dynamics of a Ship Moored by a Multi-Legged Cable System in Waves," 1972.

Brown, B. F. and R. J. Goode. "Evaluating Mechanical and Corrosion Suitability Materials," ASME Conference, ASME Paper No. 67-DE-7, May 15-18, 1967.

DDSP Instruction 9020.2. "Corrosion Control, Guidelines for," PMS 11-/201/SH: pm f Decembr 13, 1968.

"Harper Corrosion Guide." The H. M. Harper Co., Molton Grove, Ill.

Hunt, J.R. and M.D. Bellware. "Ocean Engineering Hardware Requires Copper-Nickel Alloys," The International Nickel Company, Inc.

Jerome, R. and J.L. March. Designers Guidelines for Selection and Use of Metallic Materials in Seawater Applications, General Dynamics, Electric Boat Division, December 8, 1966.

Muraoka, J.A. "The Effects of Fouling by Deep-Ocean Marine Organisms," Undersea Technology, May, 1963.

Muraoka, J.S. "Effects of Marine Organisms," Machine Design, January 18, 1968.

Reinhart, F.M. "Corrosion of Materials in Hydrospace - Part I, Irons, Steels, Cast Irons, and Steel Products," U.S. Naval Civil Engineering Laboratory, Port Hueneme, California, Technical Note N-900, July, 1967.

Reinhart, F.M. "Corrosion of Materials in Hydrospace - Part II, Nickel Alloys," U.S. Naval Civil Engineering Laboratory, Port Hueneme, California, Technical Note N-915, August, 1967.

Reinhart, F.M. "Corrosion of Materials in Hydrospace Part III, Titanium and Titanium Alloys," U.S. Naval Civil Engineering Lab, Prot Hueneme, Califoria, Technical Memo N-921, September, 1967.

Reinhart, F.M. "Corrosion of Materials in Hydrospace - Part IV, Copper and Copper Alloys," U.S. Naval Civil Engineerig Laboratory, Port Hueneme, California, Technical Note N-961, April, 1968.

Reinhart, F.M. "Deep Ocean Corrosion," Geo-Marine Technology, September, 1965.

Saroyan, J.R. "Protective Coatings, "Machine Design, January, 1968.

Uhlig, H.H. The Corrosion Handbook, John Wiley, New York, 1948.

Woodland, B.T. "Deep Submergence Metal Structues," Machine Design, January 18, 1968.

Alexander, D.C. "The Fatigue Life of Stranded Hookup Wire," Second Annual Wire and Cable Symposium, AD 656 334,1953.

Bennett, L.C. and K.E. Hofer, Jr. "Effect of Geometry on Flex Life of Stranded Wire," Fifth Annual BuWeps Symposium on Advanced Techniques for Naval Aircraft Electrical Systems, Washington, D.C., October 13-14, 1964.

Bigelow, N.R. "Development and Evaluation of a Lightweight Airframe and Hookup Wire for Aerospace Applications," Annual BuWeps Symposium on Advanced Wiring Techniques for Naval Aircraft, Washington, D.C., October 13-14, 1964.

Bryden, J.W. and P. Mitton. "Resistance of Rubber Covered Cable, Jacket, and Insulation Stacks to Weather and Artificial Aging," Fifth Annual Wire and Cable Symposium, AD 656 516, 1956.

Calhoun, F. "Submarine Cable System for the Florida Missile Test Range," Third Annual Wire and Cable Symposium, AD 656 256, 1954.

Csarella, M.J. and J. Parsons. "Cable System Under Hydrodynamic Loading," MTS Journal, July - August, 1970.

Dibbie, W.H. "The Development of Arctic Rubber Insulations and Jackets," First Annual Wire and Cable Symposium, 1952.

Eager, G.S. and S.P. Lambertson. "Mineral Insulation Wiring," Second Wire and Cable Symposium, AD 656 327, 1953.

Elmendorf, C.H. and B.C. Heezen. "Oceanographic Information for Engineering Submarine Cable Systems," The Bell System Technical Journal, September, 1957.

Hamre, H.G. "Working Voltage Classification of Insulation Wire," Seventh Annual Naval Air Systems Command Symposium on Advanced Techniques for Aircraft Electric Systems, Washington, D.C., October 11-12, 1966.

International Wire Products Corporation. "Flex Test Report -- True Concentric Versus Unilay Stranded Wire," International Wire Products Corporation, Midland Park, New Jersey, 1962.

ion,"
Ocean-
ology International, July/August, 1969.

Snoke, L.R. "Resistance of Organic Materials and Cable Structures to Marine Biological Attack," The Bell System Technical Journal, September, 1957.

Todd, G.F. "Cable Sheaths and Water Permeability," Eight Annual Wire and Cable Symposium, Asbury Park, New Jersey, AD 656 243, December 1959.

Smith, O.D. Connector Design Considerations for Hydrospace Environments, Oceanology International, November, 1970.

Spadone, D.M. Meeting on Cables, Connectors and Penetrators for Deep Sea Vehicles. Deep Submergence Systems Project Office, Bethesda, Maryland, January 15-16, 1960.

Thym, G.C. and R.A. Swan. Underwater Cable Connectors and Terminators for the Hydrostatic Pressures to 10,000 psi, Tenth Annual Wire and Cable Symposium, Asbury Park, New Jersey, November, 1961.

White, J.F. Cables and Cable onnectors, (C), NUSL Report No. 1060, Navy Underwater Sound Laboratory, New London, Connecticut, May 13-14, 1969.

Wolski, B. NR-1 Special Test for Effect of Short Circuit Fault Current on Hull Fittings, Electric Boat Division Report No. 418-69-001, January 15, 1969.

Wolski, B. NR-1 Special Test for Outboard Electrical Plug/Cable Assemblies and Junction Boxes, Two Volumes, Electric Boat Division Report No. 418-69-002, April 25, 1969.

Wolski, B. NR-1 Special Test for Electrical Hull Fittings, Junction Boxes and Associated Cable Assemblies, Two Volumes, Electric Boat Division, Report No. 481-68-010, October 22, 1968.

Haworth, R.F. Packaging Underwater Electrical/Electronic Components on Deep Submergence Vehicles, Insulation/Circuits, December, 1970.

Haworth, R.F. and J.E. Regan. Watertight Electrical Cable Penetrations for Submersibles - Past and Present, ASME Conference, ASME Paper No. 65-WA/UNT-12, November 7-11, 1965.

Haworth, R.F. and J.E. Regan. Watertight Electrical Connector for Undersea Vehicles and Components, ASME Conference, ASME Paper No. 64-WA/UNT-10, November 29 to December 4, 1964.

Johnson, E. Hermetic Seals in Plastic Bodied Connectors, 16th Annual Wire and Cable Symposium, Atlantic City, N.J., November 29 to December 1, 1967.

Klonaris, O. "Underwater Connectors," Underwater Science and Technology Journal, June, 1970.

Lenkey, J., III and W.W. Wyatt. Polyethylene Bonding to Metal for Cable Penetration of Pressure Hulls and Communications Applications, 17th Annual Wire and Cable Symposium, Atlantic City, New Jersey, December 4-6, 1968.

Miner, H.C. Design Study Report - Ballast Tank Bulkhead Cable Seals, EB Div. Report No. SPD 60-105, Contract N)bs 77007, October 31, 1960.

Miner, H.C. Final Report - Investigation, Design Development, and Testing of Shore Power Connector Fittings for Permanent Installation in Submarine Hulls, EB Div. Report No. U413-66-049, Contract NObs 90521, March 31, 1966.

Morrison, J.B. An Investigation of Cable Seals, Applied Physics Laboratories, University of Washington, March 1, 1954.

Nation, R.D. Deep Submergence Cables, Connectors and Penetrators, Nortronics Division of Northrup Corp., (DSSP Contract N00024-68-C-0217), February 21, 1967.

Nelson, A.L. "Deep Sea Electrical Connectors and Feed-Through Insulators for Packaging Electronics," Material Electronic Packaging and Production Conference, Long Beach, California, June 9, 1965.

Okleshen, E.J. "Underwater Electronic Packaging," Electrical Design News, Electronic Circuit Packaging Symposium, Fort Wayne, Indiana, August, 1960.

Sanford, H.L. Design Study Report - Phase Two - Electrical Bulkhead Connectors for Submarine Holding Bulkheads, EB Div. Report No. U413-67-202, December 29, 1967.

Sanford, H.L. Phase I Design Study Report - Electrical Bulkhead Connectors for Submarine Holding Bulkheads, EB Div. Report No. U412-66-056, Contract NObs 92442, March 31, 1966.

Sanford, H.L. Design Study Report - Watertight Electrical Plugs for Polaris Missile Harnesses on Submarines, EB Div. Report No. 413-62-096.

Sanford, H.L. and R.A. Cameron. Design Study Report-Molded DSS-3 Cable Splices for External Use on Submarines, EB Div. Report Nol. 413-62-211, December 12, 1962.

Sanford, H.L., et. al. Final Report-Watertight Deep Submergence Electrical Connectors and Hull Fitting for Submarines, EB Div. Report 413-65-185, Contract NObs 88518, October, 1965.

Aamodt, T. Seals for Electrical Equipment Under Water Pressure and Fusion of Marlex to Polyethylene by a Molding Process, Bell Telephone Laboratories, Report No. 56-131-41 of August 16, 1956.

Aamodt, T. Seals for Ocean Bottom Equipment Containers, Bell Telephone Laboratories, Report No. MM-61-21326 of February 28, 1961.

Briggs, E.M. et al. A Wet and Dry Deep Submergence Electrical Power Transmission System, Final Report Southwest Research Institute Project No. 03-25707-01 July 25, 1969.

Dowd, J.K. Design Report-Cable Seal for PQM Hydrophone, EB Div. Report NO. U411-61-091, July 1, 1961.

Dowd, J.K. Design Study Report-Pressure Proof Hermetically Sealed Coaxial Radio Frequency Hull Fittings for Submarines, EB Div. Report No. U413-62-095, Contract NObs 86068, June, 1962.

Dowd, J.K. and H.C. Miner. Design Study Report-Watertight Deep Submergency Cable Hull Penetrations Fittings for Submarines, EB Div. Report No. U413-62-097, Contract NObs 86-68, June, 1962.

Dowd, J.K. Pressure Proof Electrical Cable Hull Penetration Fittings for Submarines, EB Div. Report No. SPD-60-101, pp. 60-192, Contract NObs 7700, October 31, 1960.

Hackman, D.J. and B.R. Lower. Summary Report on a Study to Decrease Wire Breakage in Underwater Electrical Connectors, Battelle Memorial Institute, Columbus Laboratories, April 30, 1968.

Haigh, K.R. "Deep-Sea Cable-Gland System for Underwater Vehicles and Oceanographic Equipment," Proceedings, IEEE, Vol. 115, No. 1 January, 1968.

Haworth, R.F. Aluminaut Electrical Hull Fittings and Outboard Cable Connectors, January, 1966.

Haworth, R.F. Design Study Report: Hermetically Sealed Polaris Umbilical Cable Connectors, EB Div. Report No. SPD-60-107, p. 60-182, Contract NObs 77007 and 4204, November, 1960.

Haworth, R.F. Design Study Report: Watertight Hermetically Sealed Electrical Connectors for Submarines, EB Div. Report No. SPD 60-101, p. 60-194, Contract NObs 77007, October 31, 1960.

Haworth, R.F. Electrical Cabling System for the STAR III Vehicle, ASME Conference, ASME Paper No. 66-WA/UNT-11, November 27 to December 1, 1966.

Haworth, R.F. and J. J. Redding. Design Study Report: Pressure Proof Hull Fitting and DSS-3 Type Cables on An/BQQ-1 Sonar Array, SSN597, p. 59-134, Contract NObs 77007, October 23, 1959.

Development of PRD-49 Composite Tensile Strength Members, ASME No. 73-WA/Oct-14, J.D. Hightower, G.A. Wilkins, D.M. Rosencrantz, NUC, Hawaii, 11-15 November, 1973.

Engineering Analysis of Performance Factors for Subsurface Moorings in a Deep-Sea Environment, Hydro-Space Challenger Technical Note No. 6549-001, August, 1973, David B. Dillon.

A Fiber "B" Multiconductor Cable Subject to Bending and Tension, NUSC TM NO. EM-13-73, Rolf G. Kasper, Engineering Mechanics Staff.

A Structural Analysis of a Multiconductor Cable, NUSC Technical Memorandum No. EA11-23-73, A.D. Carlson, R.G. Kasper, and M.A. Tucchio, 72, also NUSC Technical Report No. 4549.

Design and Construction of Cables for Sensor Systems, Parts I and II, Sea Technology, Oct-Nov, 1973, Richard C. Swenson and Robert A. Stoltz, NUSC, New London, CT.

Study of Titanium Wire Rope Developed for Marine Applications, NRS, November 1973, NTIS No. AD-771-355.

Calculations of Stresses in Wire Rope, Wire and Wire Products 26, 766, 799 (1951).

Handbook of Vehicle Electrical Penetrators, Connectors and Harnesses for Deep Ocean Applications, July 1971, NTIS No. AD-888-281.

The Permeability and Swelling of Elastomers and Plastics at High Hydrostatic Pressures, Ocean Engineering, Vol. I, Pergamon Press, 1968, A. Lebovitz.

Sonobuoy Cable System Analysis, Tracor Document No. 024-029-01-12, .R. Sanders and Dr. M. Lowell Collier.

Haworth, R.F. Design Study Report: Watertight Hermetically Sealed Electrical Connectors for Submarines, EB Div. Report No. SPD 60-101, p. 60-194, Contract NObs 77007, October 31, 1960.

Haworth, R.F. Electrical Cabling System for the STAR III Vehicle, ASME Conference, ASME Paper No. 66-WA/UNT-11, November 27 to December 1, 1966.

Haworth, R.F. and J. J. Redding. Design Study Report: Pressure Proof Hull Fitting and DSS-3 Type Cables on An/BQQ-1 Sonar Array, SSN597, p. 59-134, Contract NObs 77007, October 23, 1959.

Development of PRD-49 Composite Tensile Strength Members, ASME No. 73-WA/Oct-14, J.D. Hightower, G.A. Wilkins, D.M. Rosencrantz, NUC, Hawaii, 11-15 November, 1973.

Engineering Analysis of Performance Factors for Subsurface Moorings in a Deep-Sea Environment, Hydro-Space Challenger Technical Note No. 6549-001, August, 1973, David B. Dillon.

A Fiber "B" Multiconductor Cable Subject to Bending and Tension, NUSC TM NO. EM-13-73, Rolf G. Kasper, Engineering Mechanics Staff.

A Structural Analysis of a Multiconductor Cable, NUSC Technical Memorandum No. EA11-23-73, A.D. Carlson, R.G. Kasper, and M.A. Tucchio, 72, also NUSC Technical Report No. 4549.

Design and Construction of Cables for Sensor Systems, Parts I and II, Sea Technology, Oct-Nov, 1973, Richard C. Swenson and Robert A. Stoltz, NUSC, New London, CT.

Study of Titanium Wire Rope Developed for Marine Applications, NRS, November 1973, NTIS No. AD-771-355.

Calculations of Stresses in Wire Rope, Wire and Wire Products 26, 766, 799 (1951).

Handbook of Vehicle Electrical Penetrators, Connectors and Harnesses for Deep Ocean Applications, July 1971, NTIS No. AD-888-281.

The Permeability and Swelling of Elastomers and Plastics at High Hydrostatic Pressures, Ocean Engineering, Vol. I, Pergamon Press, 1968, A. Lebovitz.

Sonobuoy Cable System Analysis, Tracor Document No. 024-029-01-12, .R. Sanders and Dr. M. Lowell Collier.

Determination of the Effect of Various Parameters on Wear and Fatigue of Wire Rope Used in Navy Rigging Systems, Phillip T. Gibson, C.H. Larson, and H.A. Cress, Battelle Columbus Labs., 15 March 1972, NTIS No. AD-776-993.

Workshop on Marine Wire Rope, The Catholic University of America, 11-13 August 1970, NTIS No. AD-721-373.

Load-Carrying Terminals for Armored Electric Cables, E.C. Czul, NRL, Washington, DC, 31 August 1965, NTIS No. AD-621-564.

A Study of the Causes of Wire Rope and Cable Failure in Oceanographic Service, September 1967, Robert B. Powell, All American Engineering Company, NTIS No. AD-658-871.

Handbook of Electric Cable Technology for Deep Ocean Applications, NSRDC (A), 6-54-70, November 1970, NTIS No. AD-877-774.

Rotary Shaft-Seal Handbook for Pressure Equalized, Deep Ocean Equipment, NSRDC (A), 7-573, October 1971, NTIS No. AD-889-330 (L).

Handbook of Vehicle Electrical Penetrators, Connectors, and Harnesses for Deep Ocean Applications, July 1971, NTIS No. AD-888-281.

Handbook of Fluids and Lubricants for Deep Ocean Applications, NSRDC (A), MATLAB 360, Revised 1972, NTIS No. AD-893-990.

Handbook of Fluid-Filled, Depth/Pressure Compensating Systems for Deep Ocean Applications, NSRDC (A), 27-8, April 1972, NTIS No. AD-894-795.

Handbook of Electrical and Electronic Circuit - Interrupting and Protective Devices for Deep Ocean Applications, NSRDC (A), 6-167, November 1971, NTIS No. AD-889-929.

Handbook of Underwater Imaging Systems Design, NUCTP 303, July 1972, NTIS No. AD-904-472 (L).

Handbook of Pressure-Proof Electrical Harness and Termination Technology for Deep Ocean Applications, October 1974, NTIS No. Not Assigned.

Cable Design Guidelines Based on a Bending, Tension and Torsion Study of an Electromechanical Cable, NUSC Technical Report No. 4619, Rolf G. Kasper, Engineering Mechanics Staff.

"An Economic Study of Subsea Hydraulic and Electrohydraulic Wellhead Control System" written as an Engineering Report No. 1298, July 1, 1974 by Cameron Iron Works, Inc., Payne Control Systems, Houston, Texas.

Underwater Electrical Cables and Connectors Engineered as a Single Requirement. Walsh, Don K. Marine Technology Society (MTS) Proceedings, 1966.

Dynamic Testing of Cables. Poffenberger, J.D., Cappadona, E.A., Siter, R.B. MTS Proceedings, 1966.

Natural and Synthetic Cordage in the Field of Oceanography. Brainard, Edward C., II. MTS Proceedings, 1967.

Establishing Test Parameters for Evaluation and Design of Cable and Fittings for FDS Towed Systems. Capadona, E.A., Colletti, William. MTS Proceedings, 1967.

Application of Glass-Hermetic Sealed Watertight Electrical Connectors. O'Brien, Donald G. MTS Proceedings, 1967.

Experimental Evidence on the Modes and Probable Causes of a Deep Sea Buoy Mooring Line Failure. Berteaux, H.O., Mitchell, R., Capadona, E.A., Morey, R.L. MTS Proceedings, 1968.

Thru Hull Electrical Penetrators for the Deep Submergence Rescue Vessel. Spadone, D. MTS Proceedings, 1969.

An Engineering Program to Improve the Reliability of Deep Sea Moorings. Berteaux, Henri O., Walden, Robert G. MTS Proceedings, 1970.

Undersea Cable Systems Design for the Eniwetok BMILS Installation. Bridges, Robert M. MTS Proceedings, 1970.

Integration as Applied to Undersea Cable Systems. Louzader, John C., Bridges, Robert M. MTS Proceedings, 1970.

Corrosion and Cathodic Protection of Wire Ropes in Sea Water. Lennox, T.J., Jr., Groover, R.E., Peterson, M.H.

Creep Tests on Synthetic Mooring Lines. Flessner, M.F., Pike, C.D., Weidenbaum, S.S. MTS Proceedings, 1971.

Underwater Disconnectable Connector. Tuttle, John D. MTS Proceedings, 1971.

Strength-Member Design for Underwater Cables. Nowatzki, J.A. MTS Proceedings, 1971.

Structural Requirements of Undersea Electrical Cable Terminations. Bridges, Robert M. MTS Proceedings, 1971.

Pressure Compensated Cables. Saunders, W. MTS Proceedings, 1972.

Considerations for Design and Specification of High Reliability Undersea Cables. Young, R.E. MTS Proceedings, 1972.

Methods of Measuring the Technical Behavior of Wire Rope. Milburn, D.A., Rendles, N.J. MTS Proceedings, 1972.

The Mechanical Response of an Electro-Mechanical Array Cable Subject to Dynamic Forces. Kasper, R.G. MTS Proceedings, 1973.

Verification of a Computerized Model for Subsurface Mooring Dynamics Using Full Scale Ocean Test Data. Chabbra, Narender K. MTS Proceedings, 1973.

Design and Performance of a Deep Sea Tri Moor. MTS Proceedings, 1973.

Evaluation of Kelvar-Strengthened Electro-Mechanical Cable. Gibson, Philip T., White, Frank G., Thomas, Gary L., Cross, Hobart A., Wilkins, George A. MTS Proceedings, 1973.

Computer Design of Electro-Mechanical Cables for Ocean Applications. Norvak, Gerard. MTS Proceedings, 1973.

An Airborn Sonar Cable-Design Problems and Their Solution. Bridges, Robert M. MTS Proceedings, 1973.

Power for Underwater Oil Production Systems. Briggs, Edward M. MTS Proceedings, 1973.

An Update on Recommended Techniques for Terminating Connectors to Cables. O'Brien, Donald G. MTS Proceedigs, 1973.

An Impregnated, High-Strength Organic Fiber for Oceanographic Strength Members. Berian, Albert G. MTS Proceedings, 1973.

A New Technology for Suspended Electro-Mechanical Cable and Sensor System in the Ocean. Swenson, Richard C. MTS Proceedings, 1975.

Application of the Finite Element Method to Towed Cable Dynamics. Ketchman, Jeffrey, Low, Y.K. MTS Proceedings, 1975.

Armor Designs Offer a Wide Range of Electro-Mechanical Cable Properties. Berian, Albert G., Felkel, Edward M. MTS Proceedings, 1975.

Bulkhead Connector Modification for Seawater Use Over Extended Periods. Dennison, G.N. MTS Proceedings, 1975.

Design for Neutrally Buoyant, Multi-Conductor Cables. Wilkins, George, Roe, Norman. MTS Proceedings, 1975.

Experimental Investigation of an Electro-Mechanical Swivel/Slipring Assembly. Tucket, Leroy W. MTS Proceedings, 1975.

Installation and Protection of Electrical Cables in the Surf Zone on Rock Seafloors. Valent, P.J. MTS Proceedings, 1975.

Lightweight Cables for Deep Tethered Vehicles. George A. Wilkins, Hightower, John D. Rosencranty, Donald M. MTS Proceedings, 1975.

Marine Corrosion of Selected Small Wire Ropes and Strands. Sandwith, C.J., Clark, R.C. MTS Proceedings, 1975.

Nonlinear Analysis of a Helically Armored Cable with Nonuniform Mechanical Properties in Tension and Torsion. Knapp, Ronald H. MTS Proceedings, 1975.

The Use of Kevlar for Small Diameter Electro-Mechanical Marine Cables. Holler, Roger A., Brett, John P., Bollard, Robert. MTS Proceedings, 1975.

Underwater Repair of Elecro-Mechanical Cables. Edgerton, G.A. MTS Proceedings, 1975.

Oceanic Cable Laying Telemetry and Viewing System. Kopsho, J., Schwan, H. Hydro Products. MTS Proceedings, 1975.

The Engineering, Manufacturing and Installation of Submarine Telephone Cable Systems. Schenck, Herbert H. MTS Proceedings, 1976.

Submarine Power Cables. Brinser, H.M. MTS Proceedings, 1976.

New Developments in Lightweight Electro-Mechanical Cables. Oxford, William, Galpern, Irwin. MTS Proceedings, 1976.

Analysis and Test of Torque Balanced Electro-Mechanical Mooring Cables. Christian, B. P., Nerenstein, W. MTS Proceedings, 1976.

Design of Torque-Free Cables Using a Simulation Model. Liu, F.C. MTS Proceedings, 1976.

Production and Performance of a Kevlar-Armored Deep Sea Cable. Wilkins, G.A., Gibson, P.T., Thomas, G.L. MTS Proceedings, 1976.

Oil Filled Electrical Cables External to the Pressure Hull on DSV Alvin. Hosom, D.S., WHOI. MTS Proceedings, 1976.

Compatibility of Underwater Cables and Connectors. Albert, G. MTS Proceedings, 1976.

Design and Performance of a Two-Stage Mooring for Near Surface Measurements. Bourgault, Thomas P. MTS Proceedings, 1976.

An Active Towed Body System Development. Ward-Whate, Peter M. MTS Proceedings, 1976.

Underwater Connectors and Cable Assemblies for Applications from Sea Level to 20,000 Foot Depths. Hall, J.R., Cole, J. MTS Proceedings, 1976.

Double Caged Armor for Increased Life and Reliability of Electro-Mechanical Cables. Berian, A.G., Felkel, E.M. MTS Proceedings, 1977.

Correlation of Makeup Wire Fracture Modes ad Mechanical Properties with Fatigue Life of Larger Diameter Cables. Moskowiltz, L. MTS Proceedings, 1977.

Life Evaluation of a 35KV Submarine Power Cable in a Continuous Flexing Environment. Pieroni, C.A., Fellows, B.W. MTS Proceedings, 1977.

Bend Limiters Improve Cable Performance. Swart, R.L. MTS Proceedings, 1977.

Strength and Durability Caracteristics of Ropes and Cables from Armid Fibers. Riewald, P.G., Horn, M.H., Sweben, C.H. MTS Proceedings, 1977.

Cable Terminations and Underwater Connectors. Lamborn, O.E. MTS Proceedings, 1977.

Development of Field Installable Terminations for Cables of Kevlar Aramid. Stange, W.F., Green, W.E. MTS Proceedings, 1977.

Underwater Electrical Cable and Connector Seals. Sandwith, C.J., Morrison, J., Paradis, J. MTS Proceedings, 1977.

New Mooring Design for a Telemetering Offshore Oceanographic Buoy. Higley, Paul D., Joyal, Arthur B. MTS Proceedings, 1978.

Forced Motions of a Cable Suspended from a Floating Structure. Bisplinghoff, Raymond I. MTS Proceedings, 1978.

Effects of Long-Term Tension on Kevlar Ropes; Some Preliminary Results. Bourgault, Thomas P. MTS Proceedings, 1978.

Performance/Failure Analysis of Acoustic Array Connectors and Cables After 6-10 Year of Service. Sandwith, Colin J. MTS Proceedings, 1978.

Flow-Induced Transverse Motions of a Flexible Cable Aligned with the Flow Direction. Hansen, R.J. MTS Proceedings, 1978.

The State of Technical Data on the Hydrodynamic Characteristics of Moored Array Components. Pattison, J.H., Rispin, P.P. MTS Proceedings, 1978.

Mooring Component Performance; Kevlar Mooring Lines. Fowler, G.A., Reiniger, R. MTS Proceedings, 1978.

Specifying and Using Contra-Helically Armored Cables for Maximum Life and Reliability. Berian, Albert G. MTS Proceedings, 1978.

The Use of Ethylene Propylene Diene Monomer (EPDM) Molded Connectors on An/BRA - 8 Towed Atenna Systems. Kraimer, Robert C., Orr, James F. MTS Proceedings, 1979.

CHAPTER 3

Wire Rope and E.M. Cable Lubrication

EMILE E. GRIGNARD

1.0	INTRODUCTION	3-2
2.0	WIRE LUBRICATION THEORY	3-2
	2.1 WIRE RE-LUBRICATION	3-2
	2.2 CORROSION PROTECTION	3-3
	2.3 SPLASH ZONE CORROSION	3-4
	2.4 MARINE ATMOSPHIC CORROSION	3-5
	2.5 CORROSION DURING WIRE IMMERSION	3-5
3.0	WIRE LUBRICANTS AND CORROSION PROTECTION	3-6
	3.1 INCORPORATION OF RUST PREVENTATIVE DURING MANUFACTURE	3-8
	3.2 TESTING	3-8
	3.3 RUST PREVENTATIVES	3-10
4.0	COMPATIBILITY OF LUBRICANT AND RUST PREVENTATIVE	3-11
	4.1 SELECTION OF A FIELD DRESSING	3-11
	4.2 GALVANIZING OF ROPES AND CABLES	3-12
5.0	FIELD APPLICATION OF LUBRICANTS AND RUST PREVENTATIVES	3-12
6.0	FIELD APPLICATION TECHNIQUES	3-13
	6.1 CABLE DRYING	3-13
	6.2 FIELD DRESSING APPLICATION	3-14
7.0	WIRE PRESERVATION DURING LONG TERM STORAGE	3-16
	REFERENCES	3-17

1.0 INTRODUCTION

One of the major causes of wire or cable deterioration and the reduction of its serviceable life is a lack of proper field lubrication techniques. The lack of proper re-lubrication of an operational wire is roughly equivalent to the purchase of a new automobile and then ignoring the need to add more oil as the vehicle is used; ultimately the engine lubricant is consumed and the moving metal parts begin to abrade each other until the engine fails. However, with proper re-lubrication, that engine, like a rope or cable, will last for an indefinite period of time. It is important to realize that any rope or cable is a complex mechanical system which operates in a hostile environment. Not only is it subject to both internal and external abrasion during use, but it is also prone to the corrosive effects of the sea water. Without proper re-lubrication techniques in the field, both the abrasion of the strands and corrosive effects of the sea water combine to reduce both the wire's load-carrying capacity and serviceable life.

This chapter will deal with the theory and need to practice field lubrication techniques as a defense against premature wire failure. Hopefully certain myths about wire lubrication will be dispelled and a clear picture of the importance of rope and cable lubricants established. The information presented in this chapter is not new, but appears to have been forgotten by many rope users. The intent is to re-introduce wire and cable lubrication as an important aspect of all winch and cable systems.

2.0 WIRE LUBRICATION THEORY

The lubricants applied to working ropes and cables provide a dual form of protection in that individual wires are protected from one another and the whole wire is preserved against the corrosive action of sea water. In order to understand the importance of wire and cable lubrication, it is necessary to realize that a wire, when in use, is a dynamically complex mechanical tool which is composed of numerous moving parts. As the wire passes over the sheave train, it is subjected to corrosion, bending, tension, and compressional stresses as it attempts to equalize the effects of the load it is carrying. The lubricant added to the wire during manufacturing permits this equalization to occur with a minimum of abrasion to the individual wires within each strand.

2.1 Wire Re-lubrication

The reapplication of a lubricant in the field cannot be stressed too strongly since it ensures that the friction between individual wires is reduced to a minimum; remember, each wire within a rope or cable is in constant contact with other wire along its entire length. If the user neglects to follow a

program of field lubrication, the manufacturer supplied lubricant is soon dissipated and direct metal to metal contact established between the individual wires of the rope. As the "dry" rope is used, the abrading of the individual wires reduces their metallic area and subsequently, the total load carrying capability of the wire itself. The effects of operating a dry rope versus a lubricated rope are best illustrated in the following chart taken from the Roebling Wire Rope Handbook which shows the results of cyclic testing of nonlubricated versus lubricated wire rope.

	10" Tread Dia Sheave Sheave/Rope Dia Ratio = 18	24" Tread Dia Sheave Sheave/Rope Dia Ratio = 43
Dry Rope	16,000 Bends	74,000 Bends
Lubricated	38,700 Bends	386,000 Bends

During this test series a 9/16" dia 6 x 19 wire rope was used. The results indicated that based on sheave size and lubrication, a lubricated wire will operate 2.4 times as long as a dry rope in the case of the 10" tread dia and 5.2 times as long when the tread dia is increased to 24". The results of such tests clearly indicate the benefits and the need for continued field lubrication of working wires and cables, as well as the importance of utilizing properly sized sheaves. The success of the 24" diameter sheave is due to the reduction in wire stresses due to the larger contact area the wire is exposed to and the reduction in individual wire movement within the rope.

2.2 Corrosion Protection

Protecting a wire from the corrosive effects of the salt water environment may well be the most important aspect of increased wire life. As discussed, the wire is usually delivered with some type of lubricant already in place on the wire, which acts as a corrosion prevention device as well as a lubricant. If renewed in the field, a rust preventative/lubricant can extend the useful life of the rope or cable by as much as five times the working life currently being experienced.

The full effect of corrosion damage to an unlubricated wire is virtually impossible to assess fully due to the complexity of the problem. Simultaneously a corroding wire is affected by a loss of metallic area in the individual wires due to chemical and electro-chemical action on the bare steel, the bare metallic contact areas between wires are pitted and roughened causing an uneven surface which forms stress points in individual wires, and finally the corroded contact surface inhibits the normal, smooth movement of the wires relative to one another

generating high stress concentrations, speeding corrison fatigue and crack propagation. All of these factors operate internally within the wire and may not be visible during a casual inspection other than the presence of surface or leaking rust.

2.3 Splash Zone Corrosion

The marine environment represents what is perhaps the most hostile climate a wire or cable may be required to operate in during its working life. At sea the wire is alternately subjected to the corrosive effects of the marine atmosphere and short immersions in sea water during lowerings. Combine this with frequent exposure to salt spray blowing aboard and a set of circumstances equivalent to splash zone conditions is produced.

When the wire in use is allowed to lose its factory-applied lubrication through a lack of re-lubrication at sea, it quickly loses its protective coating and becomes subject to the full corrosive effects of its environment. Figure 3-1 represents the typical corrosion rates which can be expected of bare steel in the marine enviroment. Based on the data presented in this figure, it is obvious that metals used in marine atmospheric conditions are subject to the highest rate of corrosion and, therefore, require proper field lurication techniques to be practiced if maximum wire life is to be achieved.

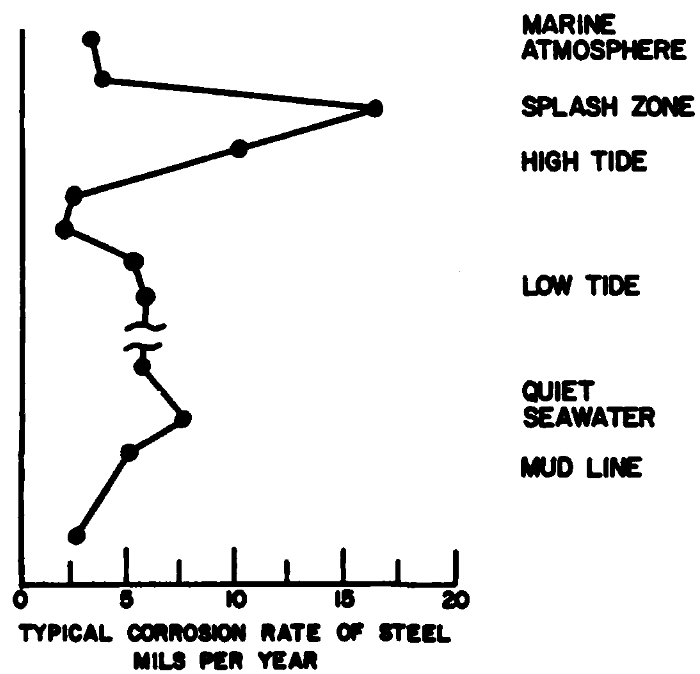


FIGURE 3-1

The effects of splash zone corrosion on an unprotected wire can be best illustrated by the following photographs. In this test performed by the International Nickle Company, equal strands of wire were exposed to a splash zone environment for three months after which time they were examined microscopically to determine the effects of splash zone corrosion on both protected and unprotected wire, show little or no deterioration of the wire's surface; while photograph 2, the unprotected wire, shows that major pitting has occurred. Based on these examples, it is obvious that a program of scheduled field lubrication is of direct benefit to wires used at sea.

The pitting that is shown in photograph 2 would be a general condition which could be expected to occur in all wires exposed to splash zone conditions. Its effect upon the unlubricated rope would be to produce a rough angular contact surface which would acclerate internal abrasion as the wires move relative to one another during use. This in turn would hasten the degradation of the rope to a point where it is neither serviceable nor safe to use. Unfortunately, this condition is virtually impossible to detect inside the wire strands and can only be protected against through a regular lubrication process.

2.4 Marine Atmospheric Corrosion

Of secondary importance to the possible mechanisms for at-sea wire degradation are the affects of the marine atmosphere on unprotected steel. Based on the data presented in Figure 3-1, atmospheric corrosion has a lesser affect upon a wire than one in the splash zone. Nonetheless, its continuing effects will be realized over time. Given the circumstances of a reel of wire stored on a vessel without adequate protection or a covering, it can be expected that at least the upper layers of the wire will be rendered useless in a relatively short period of time.

One prime example of the affects of atmospheric corrosion resulted from an experiment conducted by the Grignard Chemical Company at Kure Beach, North Carolina, in 1969. In this experiment samples of unprotected steel were exposed to the salt air environment at distances up to 800 feet from the water edge. Although corrosion was present on all samples, it was determined that samples 80 feet from the water corroded at a rate of 10% to 15% faster than the 800 foot sample. Translate this to a research vessel whose wires are never more than 20 feet from the water and this potential for rapid atmospheric corrosion is obvious.

2.5 Corrosion During Wire Immersion

The corrosive effects of sea water on wires that are immersed for long periods of time, such as buoy moorings, etc., has been an area of interest for some time. It has been observed that the rates of corrosion vary from location to

location in the ocean when the wire in use is unprotected by either a lubricant or a rust preventative. Accelerated rates of corrosion have been observed, which were ultimately traced to a combined effect of chemical levels in the water and the temperature of the sea water.

This problem was first addressed by the Grignard Chemical Company in 1969 when advanced states of corrosion were first noticed on bright steel samples from the Antigua area. With the assistance of the Woods Hole Oceanographic Institution, Oregon State University, the Halan Company and U.S. Steel, a series of experiments were conducted to determine the causes of the accelerated corrosion rate observed in the Antigua sample. Test specimens of bright steel wire, galvanized wire and electromechanical cable, supplied by U.S. Steel, were submerged in sea water for half their length at three widely spaced locations, i.e., Massachusetts, Oregon, and Antigua for a period of three months. At the conclusion of the test, all specimens were removed from the sea water and the corrosion level present in each wire was evaluated.

The analysis performed jointly by the Woods Hole Oceanogrpahic Institution and Grignard Chemical Company indicated that water temperature was the prime factors involved in the accelerated corrosion rate that had been observed. Water analysis from the three test sites were as follows.

Location	Solids (Sodium Chloride)	Temperature
Oregon	3.71% Total Solids	44° F
Massachusetts	3.61% Total Solids	40° F
Antigua	3.98% Total Solids	84° F

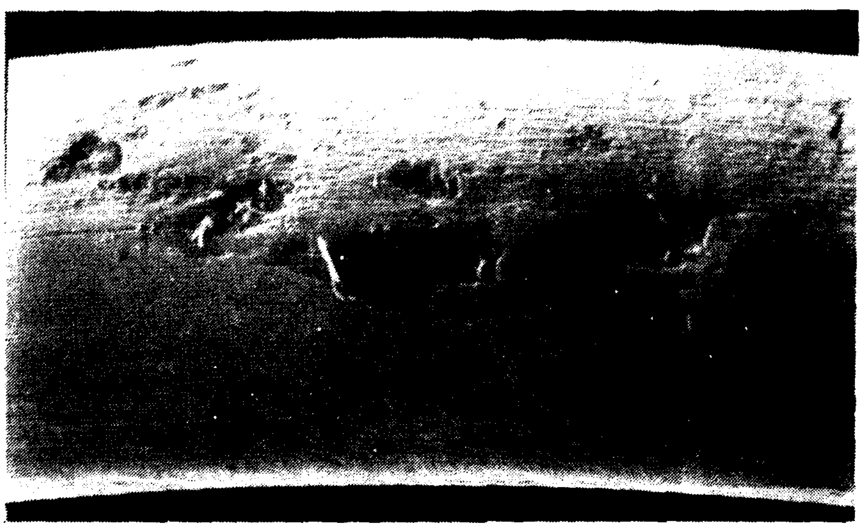
3.0 WIRE LUBRICANTS AND CORROSION PREVENTION

Essentially the function of any wire dressing is twofold in that it must act as a lubricant between individual wires to prevent premature wear and it must also prevent corrosion of the wire in the long term. Of these two functions, corrosion prevention is probably the most important as more wires fail or are discarded due to the effects of corrosion than service wear. It is a sad but true statement that oceanographic wires tend to rust out before they wear out.

Corrosion is basically a chloride reaction process where rate is increased by temperature, i.e., sodium chloride, and pollutants in the air and water. Corrosion of an unmaintained wire or cable is not restricted to only its outer surface, but instead, attacks all of the wires individually. The result is a steady reduction of the metallic area of

Wire Protected with Lubricant/Rust Preventative

Photograph 1 (100x)



Unprotected Wire

Photograph 2 (100x)

each wire and the susceptibility of the wire to corrosion fatigue during bending over a sheave.

A corroded surface, Photograph 2, is made up of a myriad of microscopic pits and craters that under loading establish failure planes in the wire strands. The bending stresses involved in working wires are a factor in fatigue failure of a wire, but when this condition is combined with active corrosion, the failure point of the wire becomes totally unpredictable. The longevity of any wire or cable can be substantially lengthened through a program of re-lubrication in the field and in the specification of wire dressings at the time of manufacture.

3.1 Incorporation of Rust Preventative During Manufacture

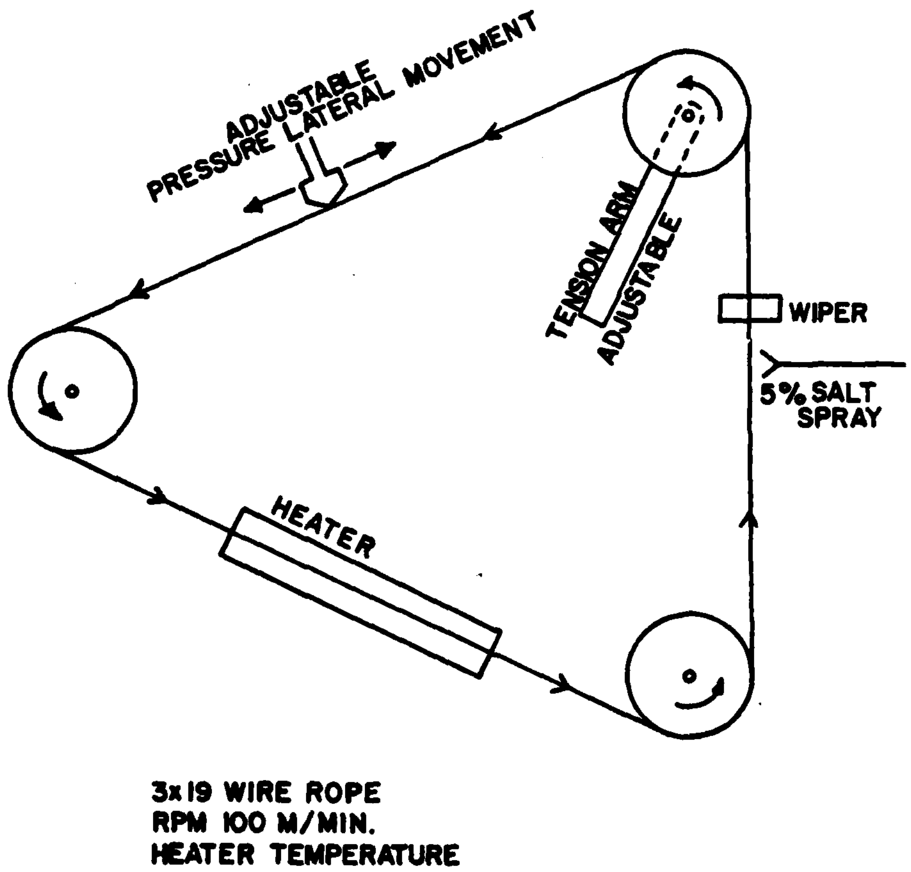
Present day manufacturing practices of applying either an asphaltic or petrolatum compound to the wire is not wholly satisfactory. Given current concerns over the longevity of ropes and cables in oceanographic service, perhaps the time is right to address new techniques for these ropes and cables at the time of their production and to develop testing procedures for the ropes. Even though most ropes and cables used in oceanographic applications are purchased on a competative price basis, it would seem that the additional cost of including an effective rust preventative would be outweighed by the greater length of service that would be obtained.

3.2 Testing

In addition, it would appear that a standard corrosion test for oceanographic ropes and cables should be developed. The current ASTM Salt Spray Panel Test is of little value where ropes and cables are concerned. In effect, the ASTM test achieves its results from monitoring the corrosion of a single steel panel in a static condition. The results of these tests confirm only that a static, high viscosity compound such as cosmoline or a shellac based product will provide good protection from corrosion. Since the hard surfaced rust preventatives are subject to cracking and flaking when the specimen is flexed, it is impossible to compare them or the ASTM test to a wire rope that is bent, stressed, and cavitates during use.

Figure 3-2 illustrates a potential test bed for oceanographic ropes and cables that could be developed to assess not only the lubricating value of compounds, but also rust preventatives. In this diagram the wire would be subjected to equivalent corrosive conditions and stresses that are present in the field. This or a similar test method would cetainly provide more relevant data than the simplified panel test.

FIGURE 3-2 PROPOSED LUBRICATION AND CORROSION TEST FOR WIRE ROPE.



3.3 Rust Preventatives

The largest problem in the effective applications of rust preventatives is being able to succesfully coat the inner wire of a rope or cable. In the manufacture of electro-mechanical cables a cotton or synthetic braid is often used between the insulated conductor and the armor which provides an ideal reservoir for a medium viscosity rust preventative (Photo 3). Although the practice of incorporating a preventative is not, at present, commonly practiced by cable manufacturers, it is a concept worthy of evaluation.

Inspections of new electro-mechanical cable that had been coated with a rust preventative after the armor was installed revealed no penetration of the preventative to the cotton braid. In use, the type of cable construction is subject to water impregnation of the braid and progressive deterioration. In a section of cable, only one and a half years old, inspection revealed that the inner armor wires were badly corroded and the braid disintegrated.

The use of an unlubricated or untreated cotton or synthetic braid can result in abrasion between the braid and the inner armor wires. In a report by the Roebling Wire Rope Company it was stated that if during manufacture the amount of lubricant is reduced too low, or is non-existant, an abrasive action between the dry fibers and the armor wires occurs as the cable is worked. This results in a form of fretting corrosion which wears extremely fine particles of steel from the armor wires which oxidize rapidly and can usually be noticed by wire discoloration. This process also results in the reduction of the ultimate strength of the cable over time.

It should be realized that a requirement, by the user, for the addition of a lubricated cotton or synthetic braid will raise the final cost of the cable by some percentage. However, this author feels that if an adequate lubricant/rust preventative is incorporated in the braid, it would be possible to extend the working life of the cable by as much as a factor of five (5) over present experience. Given this type of cable longevity, the additional cost of a lubricated braid is truly insignificant.

Wire Rope that is used in the marine environment requires an effective lubricant/rust preventative that has fluidity. Tests have shown that a medium viscosity lubricant/rust preventative applied to the strands of the rope prior to their enclosure in the final laying up of the rope has acted as an effective protective measure. Since the factory applied dressing has a finite life in a working rope, it is necessary to relubricate the rope at various times during its working life.

In order to accomplish this, a lubricant/rust preventative of medium viscosity should be used. The definition of such

a material is one which possesses a flash point no lower than 300° F (148.9° C), and a vicosity of between 300-500 SUS, (Saybold Universal Seconds), at 100° F (37.8° C). The use of solvent cutbacks should be avoided due to their low flash points and high evaporation rates which result in only partial penetration of the wire leaving the base strands open to corrosion.

4.0 COMPATIBILITY OF LUBRICANT AND RUST PREVENTATIVES

With the identified need to relubricate ropes and cables, it is appropriate to mention a few items which are of importance in achieving this process. Principally, the compatibility of the selected lubricant/rust preventative with the wire dressing provided by the manufacturer. Problems that can result from the use of incompatible materials include partial penetration of the wire, a leaching out of the components of the original compound or flaking away of the reapplied dressing due to exposure to ultraviolet light. Since these effects are not readily apparent, careful selection and evaluation of the product to be used is advised.

4.1 Selection of a Field Dressing

The solvents that are commonly used in field dressings include petroleum distillates, chlorinated hydrocarbons, diesters, glycols, and alcohols. The major problem with inexpensive petroleum solvents is their low flash point which presents a serious fire hazard when used in confined spaces or in areas where motor sparking, etc., is likely to occur. Prior to the selection of field dressing, it is advisable to contact the manufacturer and explain the intended use of the product. The fact that it is an oceanographic cable and not an elevator hoist, machinery, or automotive application may be of prime importance in obtianing an effective field dressing.

The manufacturers of available field dressings are numerous and it would be impractical to list them fully in this chapter. The following list of companies producing a field dressing is abbreviated, but regionally covers the entire country and will provide the reader with a starting point in his search for a suitable product.

COMPANY	LOCATION	TYPE
Anderson Oil	Portland, CT	Petroleum
American Oil & Supply	Newark, NJ	Petroleum & Synthetic
Ashland Oil	Ashland, KY	Petroleum & Synthetic
Citco	Tulsa, OK	Petroleum
Exxon	Houston, TX	Petroleum
Eureka Chemical	San Francico, CA	Synthetic
Grignard Chemical	Newark, NJ	Petroleum & Synthetic
Gulf Oil	Pittsburgh, PA	Petroleum

Ironside	Columbus, OH	Petroleum & Synthetic
K.S. Paul Products	London, England	Petroleum
Witco Chemical	New York, NY	Synthetic & Petroleum

4.2 Galvanizing of Ropes and Cables

It has been a standard practice in oceanographic ropes and cables to galvanize the wires prior to laying them up in final form. Zinc, used in the galvanizing process, has good resistance to sea water and usually corrodes at a rate of about 1/1000 of an inch per year, as long as the zinc cladding is intact. However, in a working oceanograhic wire internal abrasion and sheave wear can rapidly reduce or pierce the zinc on the wire.

Once the galvanized surface is broken and "white rust" is seen to form, it is indicative that degradation process has begun. In this case, the two dissimilar metals, zinc and steel, are acting against each other in the common electrolyte formed by the sea water causing an electrical flow between the two metals. One metal, the zinc coating, will become the anode while the steel acts as the cathode resulting in a steady deterioration of the galvanized coating.

It is important to remember that when the "white rust," zinc oxide, is seen, there is an abraiding action taking place in the wire which, if left alone will result in the piercing of the galvanzied coating.

The use of a lubricant/rust preventative on galvanized wire will provide the lubrication necesary to eliminate any oxide formation resulting in a longer service life of the wire.

5.0 FIELD APPLICATION OF LUBRICANTS/RUST PREVENTATIVES

Considering the hostile environment in which oceanographic ropes and cables are expected to perform, it is obvious that a program should be developed to perform regularly scheduled relubrication of wires in service. Given the current and escalating costs of ropes and cables, the small investment in time and materials required to relubricate a wire are certainly outweighed by the greater wire life which can be obtained. The majority of major wire producers recommend a relubrication procedure and fully realize the benefits to the users in the reliability of their product.

Due to the variety of conditions and use rates that exist between oceanographic organizations, it is impossible to set down a firm schedule for relubrication of working cables. The frequency of this procedure is best determined by the individual user, based on his own experience and on the wire documentation he maintains. At a minimum, the wire should be

dressed before being placed in storage ashore or every six months where actual use is on a limited basis.

For organizations with high use rates, this redressing should probably occur at two month intervals and should be applied beginning at the greatest deployment length occuring during this period. Again, the wire documentation will prove highly useful in determining where the redressing should begin. One additional consideration would be a redressing of the entire wire length prior to long periods of disuse aboard the vessel and the covering of the winch reel, where exposed to the elements, with a tarp or similar covering.

What should be remembered is that the wire or cable is an investment upon which many people depend to perform their work. Without proper care and maintenance the wire rapidly becomes unreliable and failures are subject to occur with little or no warning.

6.0 FIELD APPLICATION TECHNIQUES

In the use of field dressings it is important to remember that the majority of products on the market have a specific gravity of less than one. What this translates to in reality is that the dressing will replace moisture in and on the wire, but it will have no effect on entrained water. This brings us to the first concern in the field application of a wire dressing; the removal of excess sea water.

6.1 Cable Drying

The primary concern in preparing a rope or cable to receive a field dressing compound is that it be free from entrained water. This excess water can be removed by using a series of spaced flexible wipers constructed of either rubber, teflon or leather. It is inadvisable to use rags or other porous materials as a wiper since they quickly saturate and become ineffective. Their purpose is to scrape off the entrained water which is carried along by the moving wire. Due to the construction of 3 x 19 wire rope, a larger number of wipes are required to remove the excess water than are needed with a more concentric electro-mechanical cable.

The placement of any wiping device is relatively crucial to the success of the relubrication process. Conceptually, the wiper should be located at a point midway between the outboard sheave and the point at which the field dressing is applied. In this way some of the entrained water will naturally be shed by the wire due to vibration before reaching the wiper and the wire will have a brief opportunity to air dry before being dressed.

Under ideal circumstances it would be preferable to incorporate a compressed air dryer immediately after the wiper to further reduce the moisture level of the wire. It is realized, however, that some vessels have limitations on the availability of service air supplies which can preclude this approach. This lack of an air supply can be partially overcome by reducing the speed at which the wire is recovered during a relubrication operation.

6.2 Field Dressing Application

Essentially there are two ways of applying a wire dressing in the field; either by manual swabbing or brushing it onto the cable reel as the wire is retrieved or by a pressurized spray at the point where the wire meets the reel. In any event, the object is to completely coat the wire with the dressing and to avoid bare spots which can corrode over time. Winch speed is again of prime concern and should be kept at a speed consistant with the application technique chosen by the user.

When the field dressing is applied with a brush or mitt, swabbing, a slower winch recovery speed is required to ensure proper coverage. This process tends to be both time consuming and wastes a certain amount of the dressing compound. This technique is also the least efficient in terms of even application since a consistant pressure on the wire and dressing supply cannot be assured.

The use of a bath through which the wire passes also has certain limitations. Although there is an adequate supply of the dressing compound available, there is not adequate pressure to ensure proper coverage and penetration of the wire. In most cases the period of submergance is short and much of the compound is lost as the wire leaves the bath and is wound onto the cable reel.

The preferred method for applying a dressing compound is an automatic spray device which applies the dressing at an even pressure and volume as the wire passses. With this system individual personnel involvement is reduced to a minimum and loss of excess dressing is curtailed. The pressure behind the dressing compound allows it to penetrate the strands of the wire while excess material is trapped and recycled within the unit. A simplified sketch of an automatic spray unit is presented in Figure 3-3.

The application of field dressings tends to be an individualistic problem which is based on the facilities available to the user. The techniques which have been discussed alone will certainly accomplish the task to one degree or another and no single technique can be recommended above another as a universal approach. What should be remembered is the relative level of efficiency of each technique, as it applies to each individual case, and the fact that some gains can be achieved in

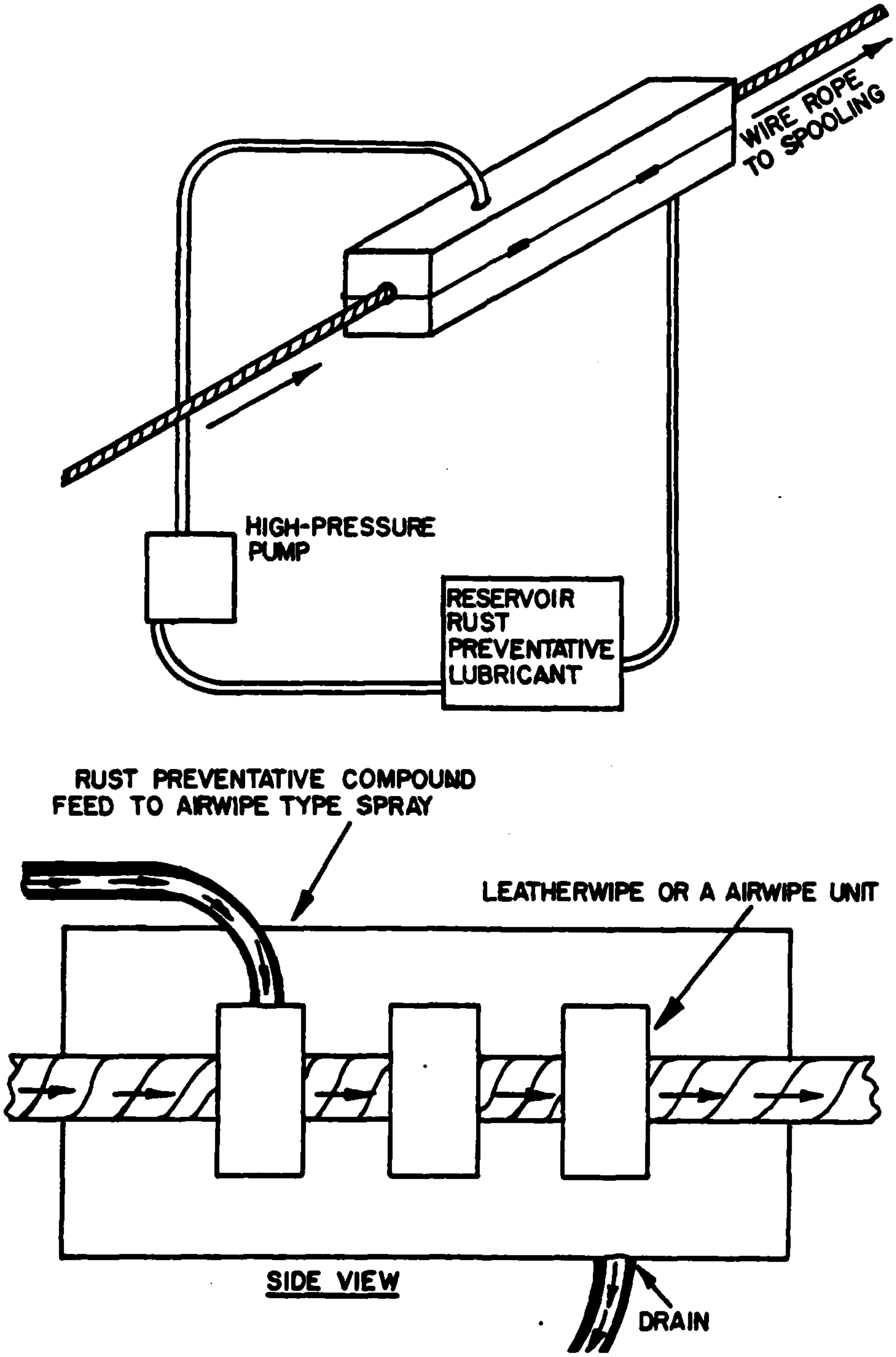


FIGURE 3-3 CONCEPTUAL AUTOMATIC SPRAY UNIT.

the swabbing or bath technique by utilizing slower winch speeds.

7.0 WIRE PRESERVATION DURING LONG TERM STORAGE

Wire rope or cable that will be stored for long periods in marine atmospheric conditions should be treated somewhat differently than the wires maintained at sea. This process involves three steps which, in actuality, occur simultaneously and involve a fresh water spray, air drying, and the application of a lubricant/rust preventative. The process can be accomplished as the wire is removed form the ship's winch to the storage reel.

The fresh water spray acts to dissolve and remove any salts that may have remained on the wire after its last use. It is preferable that the fresh water be heated prior to spraying so that a maximum dissolution of salts, etc., can be achieved. The use of a spray rather than a washing with a hose allows for a maximum effect with a minimal water volume thereby increasing the efficiency of the air drying step.

Air drying can be accomplished using a shoreside compressor and moderate pressure of approximately 80 psi. This, as in the field technique, serves to reduce the entrained moisture level and allow proper penetration of the lubricant. The lubricant/rust preventative can be applied utilizing the same unit as described in the field technique above.

Once the preservation process is completed, the reel should, ideally, be stored inside a building out of the weather as further protection. If this is not possible, a second alternative of covering the reel with a tarp is suggested. A periodic inspection and rotation of the stored reel is suggested as a means of keeping the lubricant/rust preventative evenly distributed throughout the spooled wire.

REFERENCES

Roebling Wire Rope Handbook. The Colorado Fuel and Iron Corporation, 1966.

Fink, F.W. and W.K. Boyd. The Corrosion of Metals in the Metals in the Marine Environment, Defense Metals Information Center, DMIC Report 245, 1970.

Meyers, J.J., C.H. Holm and R.F. McAllister, Ed. Handbook of Ocean and Underwater Engineering, McGraw-Hill, 1969.

CHAPTER 4

Wire Rope and Cable Operational Characteristics

PHILIP GIBSON

1.0 INTRODUCTION 4-2

2.0 REACTION OF WIRE ROPES AND CABLES TO TENSILE LOADING 4-2

2.1 SIGNIFICANT TENSION INDUCED STRESSES 4-2

2.2 WIRE ROPE AND CABLE TORQUE 4-3

2.3 CABLE HOCKLING AND KINKING 4-5

3.0 REACTION OF A WIRE ROPE OR CABLE TO BENDING 4-7

3.1 WIRE BENDING STRESSES 4-7

3.2 RELATIVE MOTIONS AMONG ROPE AND CABLE COMPONENTS 4-8

3.3 STRENGTH REDUCTION DUE TO BENDING 4-9

3.4 OTHER CONSIDERATIONS RELATIVE TO ROPE AND CABLE BENDING 4-10

4.0 ROPE AND CABLE INTERACTION WITH SYSTEMS COMPONENTS 4-14

4.1 WINDING ON DRUMS 4-14

4.2 SHEAVES 4-15

4.3 CABLE REEVING CONFIGURATIONS 4-18

5.0 WIRE ROPE AND CABLE FATIGUE DATA 4-19

6.0 ACCEPTABLE TESTS OF WIRE ROPES AND CABLES 4-22

6.1 WIRE ROPES 4-22

6.2 ELECTROMECHANICAL CABLES 4-23

1.0 INTRODUCTION

This chapter of the handbook describes the basic operational characteristics of wire ropes and electromechanical cables and the influence of the handling system design on rope and cable performance. The reaction of ropes and cables to tensile loading is explored first, including discussions of significant internal loads and stresses, tension-induced torque, and the potential for hockling and kinking.

Next, the reaction of ropes and cables to bending is described including the bending stresses and motions induced in the rope or cable components and the effects of bending on breaking strength. Also included is a discussion of the influence of sheave wrap angles and cycling stroke amplitudes on rope and cable fatigue performance.

The interaction of ropes and cables and the sheaves and drum of the handling system is next described, together with recommended reeving configurations. Examples of typical wire rope bending fatigue data are included.

Finally, recommendations are made relative to laboratory testing which is required to determine whether a rope or cable conforms to its design and/or performance specifications.

2.0 REACTION OF WIRE ROPES AND CABLES TO TENSILE LOADING

This section of Chapter 4 discusses the reaction of a wire rope or cable to straight tensile loading. A later section of this chapter explores the combined effects of tension and bending.

2.1 Significant Tension Induced Stresses

When a wire rope or cable is loaded in tension, the tensile stresses produced in the individual wires are dependent upon the helix angles of the wires within the structure. For a double armored electromechanical cable, the tensile stress experienced by the wires in one layer is related to the tensile stress experienced by the wires in the other layer by approximately the ratio of the squares of the cosines of their helix angles. Since the two wire layers typically have appoximately the same helix angle, they also experience nearly the same tensile stresses. For a wire rope, the wires in the various layers within each strand exhibit a variation in lay angle and, therefore, a corresponding variation in their tensile stresses.

The tension induced elongation exhibited by a wire rope or cable is the result of both the elastic stretch of the wire material and also slight changes in rope or cable geometry, primarily those resulting from a slight reduction in diameter. This change in rope or cable diameter is due to the radial

forces exerted by the helically wrapped wires and strands and the corresponding core compression and contact deformation at each wire-to-wire contact site.

The radial forces exerted by the outer strands of a wire rope produce quite high localized contact loading between wires of adjacent strands or between the wires of the main strands and the rope core. These high contact loads produce correspondingly high contact stresses which contribute to rope deterioration through internal wear and through the initiation and propagation of fatigue cracks.

Six-strand wire ropes are typically manufactured with sufficiently large cores to prevent wire-to-wire contact between adjacent outer strands, at least during the early life of the rope. In the case of a rope having an independent wire rope core, high contact forces and stresses are produced between the outer strands and the independent wire rope core and between adjacent strands within the independent wire rope core. These high contact loads and stresses lead to core deformations which allow the outer main strands to come into contact with each other. When this happens, high localized contact loads and stresses also develop at points of wire contact between adjacent outer strands.

In the case of the wire rope having a strand core, such as a 7 x 7 construction, each of the outer strands experiences much better support than is achieved with an independent wire rope core. An independent wire rope core has a very rough exterior surface and, as a result, provides only relatively few support points for the outer strands. A strand core, on the other hand, is relatively smooth and uniform and provides many more support points for the outer strands. As a result, the contact load and each support point is reduced, as is the contact stress.

Of course, for three-strand wire ropes, there is no core, and the three main strands maintain intimate contact at all times.

The elongation and diameter reduction experienced by rope or cable during tensile loadig also produce some bending and torsional stresses in the individual wires. However, these stresses are so small in magnitude that they can be ignored. By far, the most significant tension-induced stresses produced within rope or cable structure are the tensile stresses and the wire-to-wire contact stresses. Indeed, it is typically the contact stresses which determine the ultimate fatigue performance of the rope or cable.

2.2 Wire Rope and Cable Torque

A double armored cable, because of its contrahelical design, develops very little internal torque in reaction to

applied tensile loads. However, a six-strand wire rope which is subjected to a tension load develops a significant amount of internal torque. (It is interesting to note that although large torque magnitudes can be developed when the ends of the rope are secured to prevent rotation, the individual wires within a straight rope experience primarily tensile stresses and wire-to-wire compressive contact stresses. No significant torsional stresses are produced.)

If a wire rope is subjected to a tension load while one end is allowed to rotate, the direction of rotation will tend to loosen the outer strands and increase their lay length (the length along the axis of the rope required for a strand to complete one helical pitch). Also, the rotation of a regular lay rope tends to tighten the outer wires of each strand, whereas the rotation of a Lang lay rope tends to loosen the outer wires. This stress redistribution due to rotation effectively diminishes the metallic area of the rope which supports the applied tension load. Furthermore, the wire looseness which is produced within a Lang lay construction often leads to "secondary bending" of the outer wires if these wires are depressed against the strand surface such as when a wire rope passes over a sheave. These secondary bending stresses may contribute to premature fatigue failure of the wires.

The torque produced by a wire rope is due to both the helix of the main strands and also the helix of the wires within the strands. In a Lang lay rope, wherein the helical directions of the strands and wires are the same, the strand torque and wire torque are additive. In a regular lay rope, the torque produced by the individual wires is in the direction opposite to the torque produced by the strands. Thus, a regular lay rope has signifiantly lower torque than a Lang lay rope.

Since any rotation of a wire rope leads to a stress imbalance among the individual wires and can also lead to secondary bending within the outer wires of Lang lay constructions, the best overall wire rope performance can usually be obtained when rope rotation is prevented. Similary, within double armored cables which contain contrahelically wrapped layers of wires, any rotation will tend to increase the tensile stresses on the wires wrapped in one direction while it reduces the tensile stresses on the wires wrapped in the opposite direction. The result will be a wire stress imbalance which may contribute to accelerated cable deterioration. If two wire ropes or cables are to be attached together, their torque characteristics should be well matched to avoid rotation.

There are many potential sources of cable rotation. For example, whenever a nonsymmetrical body is towed or lowered into the ocean, rotation of the body can induce twist into the cable unless a swivel is used. Of course, if the cable itself is not of a torque balancned design, potentially damaging rotation will occur if the cable is used with a swivel. However,

even if a swivel is not used, a nontorque-balanced cable may exhibit sufficient rotation to affect its performance. This situation can occur whenever a long length of cable is suspended in the ocean.

Consider for example, a heavy nontorque-balanced cable which is suspended more or less vertically in the deep ocean. Because of cable weight, the tension near the surface will be larger than the tension near the bottom. The high tension near the surface will tend to produce a higher cable torque than will the lower tension near the bottom. However, under static conditions, there can be no torque gradient in the cable since no externally applied torsional forces exist anywhere along the cable length.

In order to seek a condition of uniform torque in the presence of a tension gradient, cable rotation will occur which will alter the inherent cable torque characteristics. The rotation near the surface will be in a direction which reduces the inherent cable torque, while the rotation near the bottom will be in a direction which increases the torque.

The extent to which these changes occur as the cable seeks a uniform torque condition is dependent upon the suspended length, the in-water weight, and the torque versus tension characteristics of the cable. Even though rotation may be prevented at each end of the cable, rotation will take place over the entire length of the cable with a maximum rotation occurring near the middle of the suspended length. During cable deployment and recovery as the suspended length and, thus, the suspended weight changes, the cable will experience a continuing change in rotation as it seeks a condition of uniform torque. Thus, whenever a cable is to be suspended or towed in great lengths, it is important that it be well torque balanced to avoid potentially damaging rotation.

2.3 Cable Hockling and Kinking

A frequent source of cable damage or complete failure is the accumulation of hockles. A hockle is a loop which forms in a cable and becomes twisted so that the portions of cable on either side of the loop become helically wrapped around each other. While the hockle itself may not seriously damage the cable, it renders the cable useless where a tension load must be transmitted to a suspended or towed body. Any application of tension to a hockled cable causes the hockles or loops to tighten, thereby producing permanent cable deformation and kinking. In a double armored cable, the outer armor wires may become badly displaced or birdcaged as a result of this hockling and kinking.

The generation of a hockle in a cable requires only that a slack loop of sufficient size be allowed to form in the presence of stored torsional energy within the cable. If a

cable contains no torsional energy, then the formation of a slack loop is not likely to produce a hockle. Similarly, if even a small amount of tension is maintained on the cable so that a slack loop cannot form, then no hockling will occur even if the cable contains a rather large amount of torsional energy.

There are a number of ways in which a cable can inadvertently form a slack loop. For example, when a package is lowered to the seafloor, especially in deep water, it may not be possible to determine exactly when the package will make contact with the bottom. If bottom contact occurs while the cable is being deployed at any significant rate, then a slack loop will immediately form at the lower end of the cable.

Whenever, a cable is used to suspend a package from a floating platform, for certain combinations of payload weight, cable deployed length, and cable elasticity, a resonant condition may develop which produces snap loading of the cable. During such snap loading, the cable alternately experiences a slack condition and then a high tension condition. During periods of slack loop formation, there is a potential for cable hockling.

In all of the above examples, some cable torque must be present in order for the slack loop to form a hockle. If the cable is designed to be nearly torque balanced over the full range of operating tensions, then hockling is unlikely to occur (assuming that no twist has been induced in the cable such as might occur during the lowering of a nonsymmetrical body). If the cable is not torque balanced, then little can be done to avoid hockling, save strict avoidance of slack loops.

If a suspended cable is attached to the payload by means of a swivel so that no torque will theoretically be developed in the cable at the lower end, cable hockling still cannot be ruled out. If the cable is not torque balanced, the swivel will undergo many revolutions as the payload is lowered toward the seafloor. At the instant of contact on the bottom, the cable tension at the lower end will suddenly drop to zero, and the swivel will attempt to spin back to allow the cable to recover a portion of the turns which accumulated during payload deployment. However, because of the rotational inertia and hydrodynamic drag of the swivel and suspended cable, it is unlikely that the swivel will be able to completely eliminate cable torque as the slack loop is rapidly forming. Thus, even the use of a swivel may not eliminate potential hockling. (And, as discussed earlier, the cable rotation which is allowed by the swivel may reduce the cable breaking strength.)

In the final analysis, then, it is quite important that a cable be well torque balanced whenever its use presents the possibility of slack loop formation during cable deployment, use, or retrieval.

Whether or not a cable actually forms a hockle depends upon a number of interacting parameters including the size of the slack loop, the magnitude of the cable torque, the torsional stiffness of the cable, and the bending stiffness of the cable. For example, a cable which has a very high bending stiffness will require a huge slack loop and a large cable torque before the slack loop will close upon itself so as to form a hockle. Conversely, a cable which has a very low bending stiffness requires only a small slack loop and a small amount of cable torque to generate a hockle.

3.0 REACTION OF A WIRE ROPE OR CABLE TO BENDING

Bending of a wire rope or cable occurs as the result of passage over sheaves, drums, or deflection rollers. Bending also occurs at a sheave tangent point (in a plane perpendicular to the sheave) whenever a fleet angle exists. To a lesser degree, bending is produced by the transverse vibrations which are common to many cable systems.

3.1 Wire Bending Stress

The bending of a wire rope or cable produces bending stresses in the individual wires due to a change in the radius of curvature of these wires. Unlike a simple beam in bending, the maximum bending stresses which occur in a wire rope are not in the wires which are furthest from the center of curvature of the rope. In fact, the maximum wire bending stresses actually occur in wires which are located nearest the rope core. If a wire rope is subjected to repeated bending at a relatively small radius of curvature and in the absence of significant tension loading, the individual wires will ultimately fail in bending fatigue in the interior of the rope. In fact, the insidious nature of such fatigue deterioration has led to accidents involving the loss of life.

However, in a great majority of common wire rope applications, the locations of fatigue failures of individual wires are either on the crowns of the strands where the wires make contact with a sheave throat, or at the locations of wire notching between adjacent strands. These fatigue failure sites often prompt the conclusion that the most severe bending stresses occur at these locations in the rope. However, the rope wires fail at these locations because of high contact stresses and wear which have a greater influence on wire fatigue life than do the wire bending stresses per se.

Within a double armored electromechanical cable, the bending stresses in the individual wires are greatest at the locations where the wires are nearest to and furthest from the center of curvature of the cable. For large sheave-to-cable diameter ratios, the bending stresses at these two locations are nearly identical.

The bending of a wire rope or cable also produced torsional stresses in the individual wires. However, analysis has shown that these torsional stresses are quite small relative to the bending, tension, and wire-to-wire compressive contact stresses which control the ultimate fatigue life of the rope or cable.

3.2 Relative Motions Among Rope and Cable Components

Another phenomenon which takes place as a rope or cable is bent is relative motion among the strands and wires. The magnitude of this relative motion diminishes with increasing sheave-to-cable diamteter ratios. Since wire ropes and cables are far from frictionless, especially when subjected to a significant tensile load, the effect of the internal friction is to reduce the magnitude of the relative motions among the various components. In the limit, if a rope or cable were totally frictionally stiff such that no motion at all occurred among its components, then it would behave like a simple beam in bending.

The internal friction which exists in all wire ropes gives rise to changes in the stress distributions among the rope components when the rope is bent. Because the individual strands are inhibited from moving, the load distribution among the strands does not remain uniform. Rather, in the vicinity of a sheave tangent point, some strands carry a greater proportion and others carry a lesser proportion of the rope tension then they would in the absence of the sheave. This redistribution of strand loading causes the rope to develop a slight "corkscrew" appearance in the immediate vicinity of the sheave tangent point. This phenomenon can be readily observed when large wire ropes are caused to bend over relativley small diameter sheaves.

Thus, the bending of a rope produces not only bending stresses in the individual wires, but also relative strand motions which lead to wire-to-wire wear within the rope structure and which also produce variations in the tensile loading among the strands due to the inherent rope friction. Of course, the bearing pressure produced in the zone of contact with a sheave further aggravates the internal friction, contact loading, and contact stresses. In this case, the rope not only experiences increased internal loads and stresses, but also wear and high contact stresses at the locations where the outer wires contact a sheave. Of course, all of these effects are diminished with increasing sheave sizes.

The magnitude of the rope distortion and the alteration in the load distribution among the strands which results from rope bending is, of course, a function of the internal rope friction. A rope which is well lubricated will have lower internal friction and, as a result, will experience smaller variations in strand loading than will a less well lubricated rope. Furthermore, a rope which suffers from internal corrosion will exhibit the greatest variation in individual strand ten-

sions as a result of rope bending. These variations in strand tension produce variations in tensile stresses within individual wires which, when added to the bending and contact stresses, lead to accelerated fatigue damage.

Within any rope or cable structure, the magnitude of the relative motions between adjacent elements in a given layer of strands or wires diminished with increasing numbers of individual elements. Thus, the relative motion between any two adjacent wires in a given layer within a double armored cable is quite small as compared to the relative motions between any two adjacent strands in a wire rope.

However, even though increasing numbers of elements within a given helical layer diminishes the relative motions between any two adjacent elements in that layer, nevertheless, the relative motions which take place between adjacent layers remain quite large if these layers have opposite helical directions. For a double armored cable, the result is wire wear between layers in addition to the high contact stresses previously discussed.

3.3 Strength Reduction Due to Bending

The maximum tension experienced by a rope or cable usually occurs where the rope passes over the sheaves and drums in the handling system. Any time a rope is bent over a sheave or drum, its breaking strength is diminished. The smaller the sheave-to-rope diameter ratio, the lower will be the effective breaking strength. Approximate strength efficiencies achievable for 6 x 19 and 6 x 37 class ropes are shown in Figure 4-1.

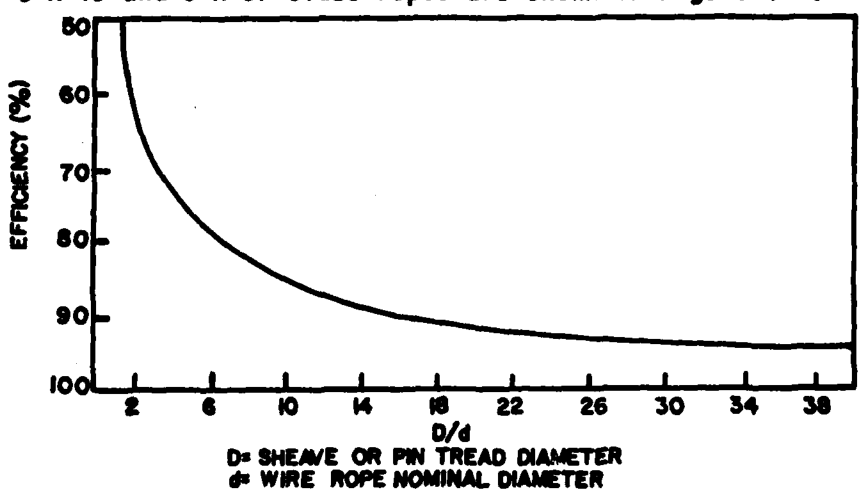


FIGURE 4-1

APPROXIMATE STRENGTH EFFICIENCY OF WIRE ROPE WHEN BENT OVER SHEAVES OR PINS OF VARIOUS SIZES.

This curve applies to a "static" condition wherein the rope is not moving relative to the sheave pin. In this case, the rope initially experiences bending while being subjected to essentially zero tension, thus allowing the corresponding relative motions to take place among the rope components without being significantly inhibited by internal rope friction. When the rope is then pulled to failure, the attainable breaking strength is influenced primarily by the compressive contact stresses within the rope and between the rope and the sheave or pin (although the bending stresses and some nonuniform distribution of tensile stresses among the strands also play a minor role).

An even greater strength reduction is experienced if the rope is moving over the sheave. In this case, the relative motions among the rope components in the vicinity of the sheave are inhibited by the extreme, tension-induced internal rope friction. This condition produces a nonuniform tension distribution among the strands. The most highly loaded strand(s) then fail prematurely and produce a reduced rope breaking strength.

In the case of double-armored electromechanical cables, similar strength reductions are experienced. The amount of this strength reduction is dependent upon the sheave-to-cable diameter ratio and on the details of the cable design. Usually, cables having relatively small helix angles for the armor wires will exhibit a greater strength loss due to bending.

3.4 Other Considerations Relative to Rope and Cable Bending

Consider a wire rope which moves over a sheave with a large stroke amplitude such as during deployment and retrieval. Experiments have shown that if the arc of contact between the wire rope and a sheave encompasses at least one lay length of the strands, then the bending fatigue life of the rope is not a function of the rope wrap angle around the sheave. This behavior is shown graphically in Figure 4-2. For a certain range of contact arcs less than one rope lay length, the bending fatigue life of the rope may actually be reduced depending on the rope construction, the sheave-to-rope diameter ratio, and the operating design factor. (The design factor equals the rope breaking strength divided by the operating tension.)

Figure 4-3 shows the rope wrap angles on a sheave required to produce arcs of contact equal to one-half or one rope lay length for various sheave-to-rope diameter ratios. This graph assumes that the rope lay equals 6.7 times the nominal rope diameter, a value which is typical for so-called "working" ropes of a six-strand construction.

When a wire rope passes over a sheave under tension, all changes in wire stresses due to bending and due to the relative motions among the rope components take place within ap-

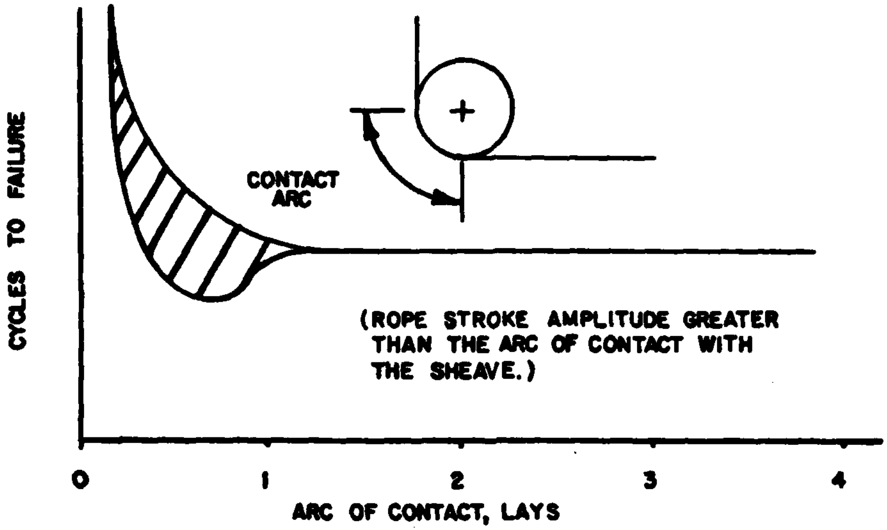


FIGURE 4-2. EFFECT OF SHEAVE CONTACT ARC ON WIRE ROPE BENDING FATIGUE LIFE.

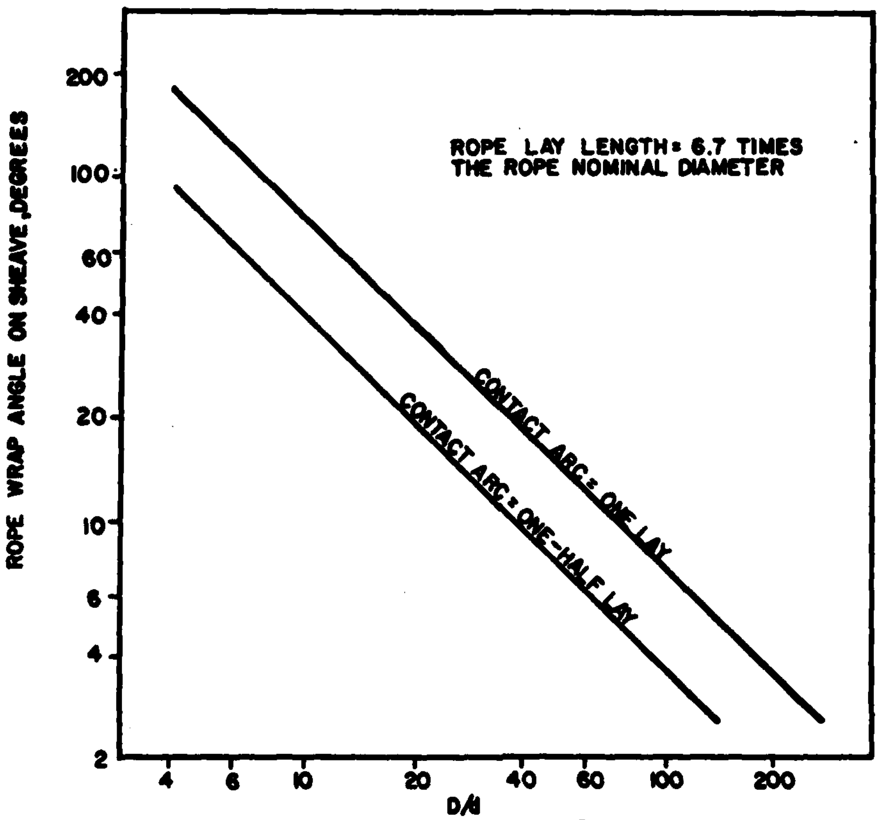


FIGURE 4-3 WRAP ANGLE OF A WIRE ROPE ON A SHEAVE CORRESPONDING TO VARIOUS CONTACT ARCS

proximately one lay length either side of each sheave tangent point. This behavior has been confirmed in the laboratory by monitoring strain gauges installed on individual rope wires. If the arc of contact exceeds one lay length, there will be a certain portion of the rope in contact with the sheave which, having undergone stress changes in the vicinity of one sheave tangent point, will experience no further changes in its state of stress until it again approaches the second sheave tangent point. Thus, the actual length of contact between the rope and sheave has no influence on the rope stress variations or on the rope fatigue life. (Again, these comments apply to a wire rope which is moving over a sheave with a large stroke amplitude.)

The fact that the rope bending fatigue life may be diminished for a range of contact arcs less than one rope lay implies that when small arcs of contact are to be employed, it may not be advisable to arbitrarily reduce the diameter of the sheave or replace sheaves with a series of small rollers. Unfortunately, many such substitutions are made because of a misunderstanding of rope behavior.

The effect of cycling stroke amplitude on rope bending fatigue life is shown in Figure 4-4. For stroke amplitudes exceeding one rope lay length, but less than the rope-to-sheave contact arc, the wire rope bending fatigue life is not a function of stroke amplitude. For stroke amplitudes less than one rope lay length, the fatigue life is greatly increased. For stroke amplitudes greater than the rope-to-sheave contact arc, the bending fatigue life of the rope as expressed in terms of stroke cycles diminishes to one-half of that experienced with a somewhat shorter stroke amplitude. In the latter case, each long-stroke cycle produces two straight-bent-straight bending cycles of the wire rope, whereas for shorter stroke amplitudes, each stroke cycle produces one rope bending cycle.

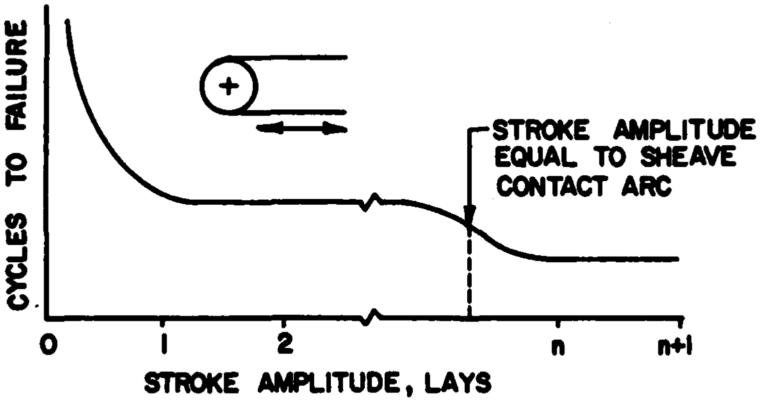


FIG.4—4 EFFECT OF CYCLING STROKE AMPLITUDE ON ROPE OR CABLE BENDING FATIGUE LIFE

Consider the interaction between a deployed wire rope and the outboard sheave. Depending on the prevailing sea conditions, there will be some flexing of the rope as it changes its arc of contact with the sheave as shown in Figure 4-5. When this change of contact arc affects only a fraction of a lay length of the wire rope, then very little rope degradation will be experienced due to bending fatigue. Only when the change of contact arc approaches one lay length do bending fatigue considerations become important.

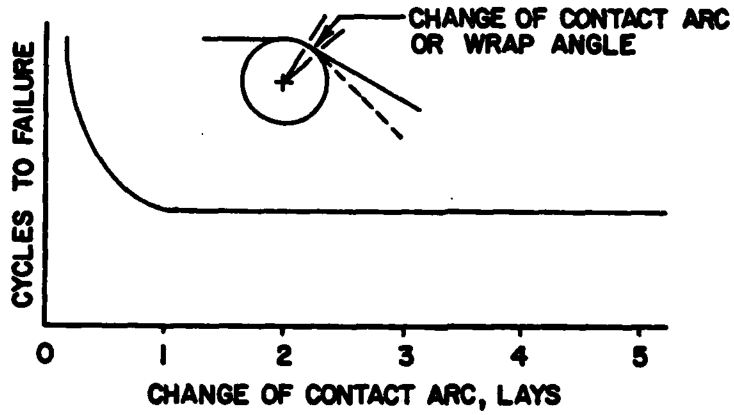


FIG.4–5 EFFECT OF BENDING AMPLITUDE AT OUTBOARD SHEAVE ON ROPE OR CABLE BENDING FATIGUE LIFE

Figure 3 may be used to determine the change of wrap angle on a sheave corresponding to a change of contact arc equivalent to one lay length of an outer strand in a wire rope. For example, for a bending diameter equal to 40 times the rope diameter, an 18-degree change of wrap angle would be required to affect one full lay length of a strand in a wire rope. A slightly greater change of wrap angle would be required to affect one full lay length of an outer wire in an electromechanical cable. These data indicate that if changes of the wrap angle on a sheave are limited to just a few degrees, then only a small fraction of a lay length will be involved in the bending, and the rope or cable should suffer little deterioration due to bending fatigue per se.

4.0 ROPE AND CABLE INTERACTION WITH SYSTEM COMPONENTS

The previous sections of this chapter discussed the benefits of large bending diameters with regard to the internal stresses, motions, and fatigue performance exhibited by a wire rope or cable. The following section describes other system design considerations which influence the useful service life of the rope or cable.

4.1 Winding on Drums

For those systems in which the rope or cable tension is always quite low at the drum (system employing some type of a traction unit), the cable life will probably not be influenced to a great extent by the details of the drum design. However, for those systems which require the cable to be wrapped on the drum under high tensions, the drum can be a major source of cable damage.

The factors which affect cable life are the drum diameter, the number of cable layers, the type of grooving, and the uniformity of winding. The influence of these factors can vary from one installation to another. However, in general, the cable life will benefit from a large diameter drum (to minimize the number of cable layers), proper grooving, and the use of a level wind system to achieve smooth winding.

If a cable must be wound on a drum in multiple layers and must also sustain a significant tension load, cable damage may occur due to crushing of the bottom layers on the drum, localized pinching and bending due to uneven winding at the drum flanges, or "cutting in" where the outer wrap of cable becomes buried within the inner wraps due to the application of a high tension load. Crushing of the inner wraps of cable can be minimized by using large diameter drums which will cause the cable to develop a lower radial pressure for each layer and will minimize the required number of layers. The drum should also be properly grooved. Localized cable damage at the drum flanges can be reduced by the use of riser and filler strips. The potential for cutting in of a cable on a drum can be reduced if the winding tension is nearly the same for all layers.

Proper winding of the cable will usually require that some type of level wind system be used. In such a system, the rope is guided by a set of sheaves or rollers to assure that it is wound smoothly on the drum. These guide sheaves or rollers traverse the arc of the drum and may be driven by a mechanism which synchronizes their motion with the rotation of the drum. Care must be taken in the design of the level wind sheaves or rollers to assure that they are not a source of premature cable damage.

If no level wind system is used, the first fixed sheave away from the drum must be positioned so as to limit the cable

fleet angles (usually to less than 1-1/2 degrees). Excessive fleet angles can produce misspooling on the drum and cable wear from contact with adjacent drum wraps or sheave flanges.

In the least complicated system, the cable is wound directly on the drum under full operating tension. This type of system can be satisfactory if the cable tensions are not excessive and if there is not a large number of cable layers on the drum. Otherwise, it is likely that the cable will experience the types of damage mentioned above.

To allow the cable tension to be maintained at a low value on the drum, some type of traction system is required. While a single flat-faced or slightly tapered capstan drum may be used to control the tension of a wire rope, such a system is not recommended for an electromechanical cable. Bending of such a cable at tension should occur only on properly grooved sheaves and drums. Thus, a double-drum traction unit is often used as discussed elsewhere in this handbook. In such a system, the cable passes over a pair of grooved drums in a series of half wraps. One or both of the drums is driven electrically or hydraulically. The friction between the cable and the drum grooves allows a significant tension gradient to be developed within the traction unit, thereby allowing the cable to be wound on the storage drum at low tension.

4.2 Sheaves

During the selection of the sheaves to be used in a cable handling system, it is frequently necessary to compromise between the size and weight requirements and the desire for large sheave diameters to assure good cable flexure life. The sheave diameter which is satisfactory for a given application depends on the details of cable design, the operating cable tension, the desired number of flexure cycles, and the consequence of a cable failure. It is generally not possible to specify the minimum sheave-to-cable diameter ratio which will assure satisfactory cable performance. Some cable designs and operational requirements demand quite large sheaves, while other combinations of parameters allow much smaller sheaves to be used. However, in general, the larger the sheaves the better.

Figure 4-6 shows the various factors which must be considered during the design of a sheave. Of primary importance is the shape of the sheave groove to assure good cable support. The groove radius should be close to, but not more than five percent greater than, the cable radius as measured at zero tension. Sheave grooves which are too small will pinch the cable and cause premature failure. Sheave grooves which are too large will allow a flattening of the cable under high loads which will also accelerate cable failure. A cable should never be used on sheaves which are grooved to accommodate a cable of larger diameter.

A flange angle of 30 degrees as measured between the sheave flanges is typical for most applications. This sheave configuration will provide an adequate cable-to-sheave contact area and will preclude cable wear against the sheave flanges, assuming that the sheaves are properly aligned to avoid large fleet angles. However, in some situations it is not possible to avoid a large cable fleet angle, so a larger flange angle will be necessary. An example is an outboard sheave where large relative motions must be accommodated between a deployed cable and the support ship. In such a case, a large flange angle may be necessary to avoid cable contact with the edge of the sheave flange.

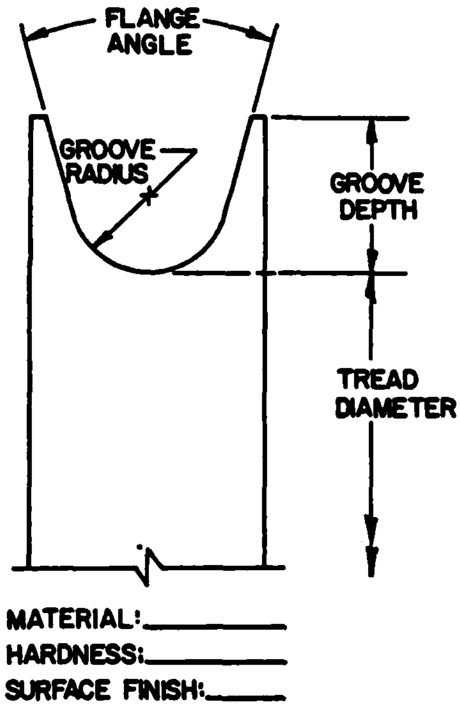


FIG. 4-6 SHEAVE PARAMETERS

A sheave groove depth of one cable diameter is usually satisfactory for many applications. However, deeper grooves may be used to assist in reeving the cable through the system or to assure that the cable does not come out of the grooves during extreme operating conditions. Also, it should be noted that whenever a cable contacts a sheave with even a small fleet angle, special attention must be paid to both the sheave groove depth and the flange angle to avoid contact between the cable and the edge of the sheave flange. For example, a sheave flange half angle of 15 degrees does not imply that the sheave will perform satisfactorily with fleet angles up to 15 degrees. The actual cable fleet angle which can be accommodated with such a sheave is a function of both the sheave groove depth and the sheave-to-cable diameter ratio. The larger the D/d ratio, the greater will be the required groove depth to avoid contact between the cable and the edge of the sheave flange.

Whenever a metal sheave is used, the entire sheave groove must have a smooth finish to avoid premature cable wear and fatigue damage, and the groove must be hardened to avoid wear and corrugation of the sheave surface. Such distortion of the sheave groove will shorten the cable service life.

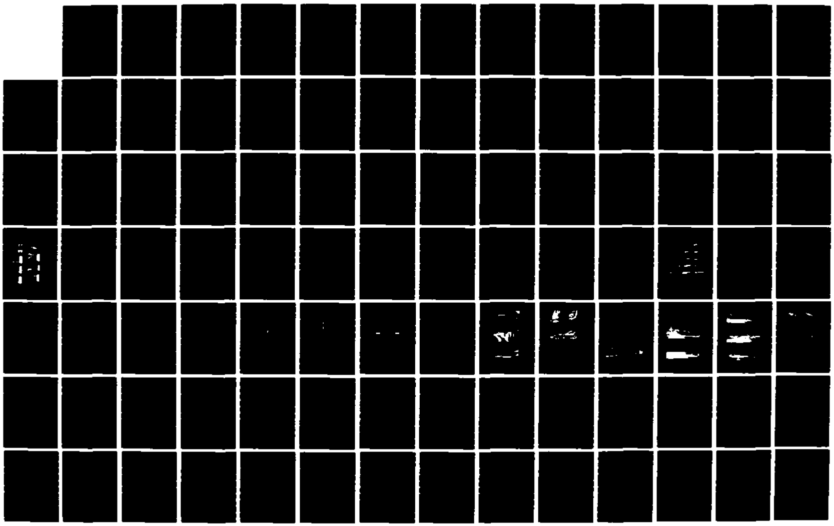
4.3 Cable Reeving Configurations

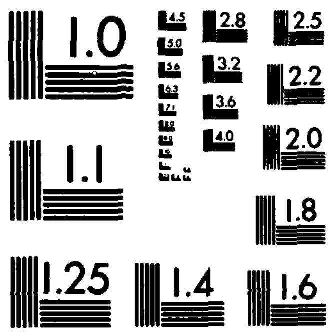
In the simplest system, the cable is deployed directly from its storage drum without passing over any sheaves or through any guide rollers. Other systems require relatively complex reeving configurations where the cable must pass over a number of sheaves. Regardless of the complexity of the system, safety considerations should not be neglected. Whenever possible, personnel walkways should be designated away from the cable system to avoid having the cable pass near or through a commonly used walkway. An equally important consideration is the recoil path the cable will have in the event of a catastrophic failure. Such cable recoil can inflict serious damage and injury at locations well away from the normal route of the cable. Where necessary, barricades should be erected to absorb the energy of a recoiling cable.

Cable systems vary not only in their complexity, but also in their frequency of use. Some systems require a cable to be deployed and retrieved relatively infrequently, while other systems may subject the cable to nearly continuous load and flexure cycling. In any case, there are a number of system design guidelines which will improve the cable service life.

Of course, the number of sheaves in the system should be kept to a minimum whenever cable flexure life is a concern. Also, the greater the number of desired cable flexure cables, the greater will be the required sheave diameters and operating safety factors.

AD-A132 513 HANDBOOK OF OCEANOGRAPHIC WINCH WIRE AND CABLE 3/5
TECHNOLOGY(U) RHODE ISLAND UNIV KINGSTON GRADUATE
SCHOOL OF OCEANOGRAPHY A H DRISCOLL 18 OCT 82
UNCLASSIFIED N00014-82-G-0116 F/G 8/10 NL





MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

All components of the cable handling system should be arranged so as to minimize the cable fleet angles. As discussed earlier, large fleet angles can lead to misspooling of the cable on the storage drum and cable wear due to rubbing on adjacent wraps on the drum or on sheave flanges. Large fleet angles are also detrimental because they produce a small radius cable bend at each sheave flange in a plane perpendicular to that of the sheave. Such small radius bends are potentially as damaging to the cable as small radius sheaves.

Whenever possible, the cable routing from the drum and through the various sheaves should be chosen so as to eliminate reverse bending of the cable. A cable which is bent in the same direction over two sheaves will have considerably better service life than if it is subjected to a reverse bend over the same two sheaves. While it is sometimes impossible to avoid reverse bending of the cable, the consequences of this reeving configuration must be recognized.

It is a common misconception that a cable can be routed in an arc over a series of small diameter rollers without affecting the performace which would otherwise be obtained by passing the cable over a sheave having the same pitch radius. While such an arrangement of small rollers may be acceptable if the cable tension is maintained at near zero, these same rollers will severely damage the cable at normal operating tensions. Each roller will subject the cable to a severe bending condition, even though the cable wrap angle over the roller is quite small. Thus, it is always advantageous to eliminate guide rollers whenever possible in favor of sheaves having the proper geometry.

Finally, it is quite important that a moving rope or cable not come into contact with any stationary structure, since the resulting frictional heating of the surface of the wires can easily produce a thin layer of untempered martensite. This martensite is very hard and brittle and will develop small cracks as soon as the rope or cable is subjected to any significant tension or bending. Experience has shown that these cracks then propagate rapidly through the remainder of the cross section and produce premature wire failures.

5.0 WIRE ROPE AND CABLE FATIGUE DATA

As expected, cyclic bending fatigue data indicate that improved performance can be achieved with larger bending diameters and larger safety factors (lower operating tensions). However, it is not possible to identify a specific bending diameter or a specific safety factor which is optimum for a given wire rope or cable. Simply stated, the more severe the operating conditions in terms of bending diameter and tension load, the shorter will be the achievable service life.

Figure 4-7 shows typical bending fatigue data for common 1/2-inch diameter wire ropes. In this figure, the rope bending cycles to failure are plotted as a function of a dimensionless parameter called the "life factor." This parameter is simply the product of the design factor (the rope breaking strength divided by the test tension) and the ratio of the sheave-to-rope diameters. In this case, the sheave-to-rope diameter ratio is based on the pitch diameter of the sheave as opposed to the tread diameter. These data points represent all possible combinations of the specified rope size, rope construction, and test tension.

For the purpose of this discussion, the conventional definition of a rope bending cycle is used; namely, the straight-bent-straight flexure of the rope specimen. (For a cycling stroke amplitude of less than one-half sheave circumference, each cycle of the test machine produces one rope bending cycle. For stroke amplitudes in excess of one-half sheave circumference, each machine cycle produces two rope bending cycles.)

The life factor parameter allows a fairly good normalization of the fatigue data for a wide range of test parameters. While other methods are often used for plotting wire rope fatigue data, the life factor allows an analysis of rope performance based on the familiar parameters of design factor and sheave-to-rope diameter ratio.

It should be noted that the data presented in Figure 7 apply to a specific set of wire ropes. Wire ropes of the same size and construction from various manufacturers and with various material strengths will have somewhat different fatigue characteristics. Also, larger diameter wire ropes will typically exhibit poorer fatigue lives. However, the curve in Figure 7 is useful for estimating how the fatigue performance of a wire rope will be affected by changes in sheave sizes or design factors.

For double armored electromechanical cables, no generalized bending fatigue data are yet available for a range of sheave sizes and designs factors. However, the data which do exist suggest that double armored cables should not be operated at life factors less than approximately 60; extreme data scatter is encountered with more severe conditions. Furthermore, for life factors greater than 60, the number of bending cycles required to produce the first broken outer armor wire is typically greater than the number of cycles required to produce complete failure of a wire rope operating at the same life factor. Thus, for life factors greater than 60, Figure 7 can also be used to estimate the mechanical flexure life of double armored cables. (Of course, the number of cycles which a cable will survive prior to electrical failure may be much smaller.)

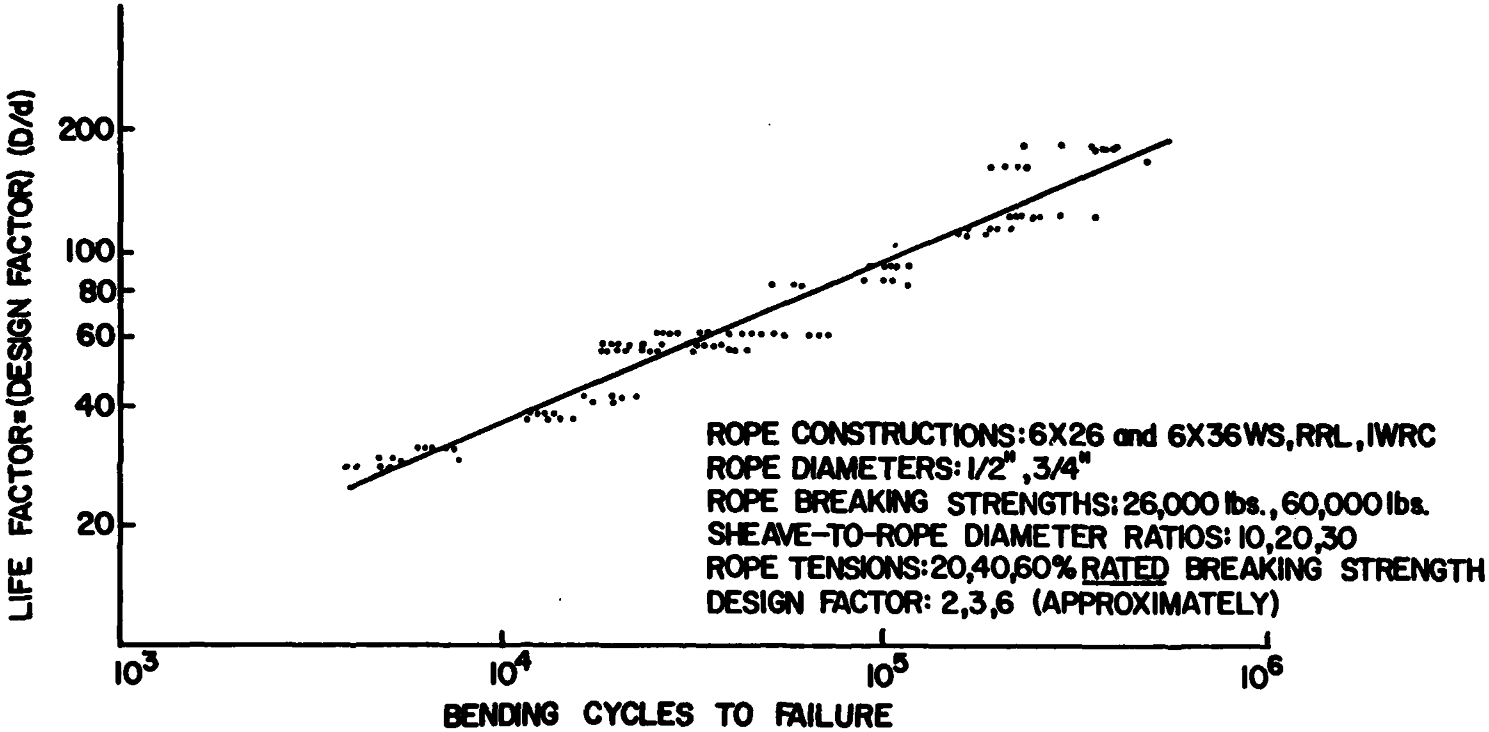


FIG. 4-7 WIRE ROPE BENDING FATIGUE LIFE

6.0 ACCEPTANCE TESTS OF WIRE ROPES AND CABLES

To assure optimum performance of a wire rope or electromechanical cable, certain tests are required to confirm that the rope or cable components, as well as the complete assembly, conform to applicable design or performance specifications. A majority of these tests will be a part of the normal manufacturing quality assurance procedures. However, more thorough tests which are directly related to the end use of the wire rope or cable are usually required to quantify the useful service life, to identify the modes of failure, or to identify constructions or manufacturers which provide the best overall performance.

6.1 Wire Ropes

Commonly available wire rope constructions are typically of high quality. Rope manufacturers routinely conduct tension and torsion tests and chemical analyses on the individual rope wires to confirm their suitability for the intended rope construction. Breaking strength certificates for the complete rope are also available if requested when the rope is initially ordered.

To gain additional assurance of high wire rope quality, the rope procurement document should reference appropriate specifications such as Specification 9A for Wire Rope published by the American Petroleum Institute. This specification fully describes the physical properties of the individual rope wires, the rope design details, and the physical properties of the complete rope.

Of course, any wire rope should also be inspected when it is received, perferably during installation on the winch drum. The main purpose of this examiantion is to confirm uniformity of rope geometry and diameter and to identify potential damage which may have occurred during rope manufacture or shipment. This inspection, combined with a certificate from the manufacturer indicating conformance with the applicable specifications and measurement of the acutal rope breaking strength, should be sufficient to confirm product quality.

However, more detailed laboratory testing is required in the interest of identifying the optimum rope construction for a given application, for comparing the products of various manufacturers, for identifying the rope failure modes, and for developing suitable rope retirement criteria. For example, cyclic-bend-over-sheave fatigue tests can be conducted in the laboratory using sheave geometries and tensions which are representative of the operational system. The development of fatigue curves such as shown in Figure 7 will allow the optimum rope to be identified and will reveal how rope performance can be altered by changes in the system geometry or operating tensions.

6.2 Elecromechanical Cables

Unlike wire ropes, electromechanical cables are available in a wide variety of configurations which reflect the differing requirements for both the electrical conductors and the armor package. This variety of construction details, combined with a broad spectrum of operational requirements, has hampered the development of generalized specifications for such cables. As a result, it becomes the responsibility of the cable user to provide the cable manufacturer with complete details of the cable operational requirements so that appropriate cable materials and construction details can be selected.

The cable procurement document must specify critical parameters such as the electrical performance requirements, the minimum breaking strength, and the required torque and rotation characteristics. The cable manufacturer should be required to confirm proper cable performance through his routine quality assurance test program. However, special laboratory tests will be required to determine how well a specific cable will perform for a given application. Typical laboratory tests include cyclic ending fatigue, cyclic tension fatigue, vibration or strumming, hydrostatic pressure, elongation versus tension, torque versus tension, rotation versus tension, and flexing of the cable at a termination. If properly conducted, these tests will reveal how long a given cable may be used in a specific system before electrical failures are likely to occur or before the safety of the system is jeopardized by general cable wear and fatigue deterioration.

Such laboratory tests are essential for any cable which is used with a manned system or which is used to deploy expensive instrumentation packages or for any system where the cost of the cable itself is quite high due to cable complexity or length. A modest laboratory test program will reveal the operational limitations of a specific cable and will significantly reduce the chances of a costly cable failure in service.

CHAPTER 5

Wire and Winch Documentation

ALAN H. DRISCOLL

1.0 INTRODUCTION	5-2
2.0 WIRE AND CABLE DOCUMENTATION	5-3
2.1 WIRE AND CABLE IDENTIFICATION	5-3
2.2 WIRE SPECIFICATIONS	5-4
2.3 WIRE DOCUMENTATION SYSTEM	5-4
3.0 WIRE USE RECORD	5-7
4.0 WIRE INVENTORY RECORD	5-9
5.0 SHORE BASE WIRE LOG	5-11
6.0 WIRE FAILURE REPORT	5-12
6.1 SHIPBOARD ANALYSIS	5-15
6.2 ENGINEERING ANALYSIS	5-17
6.3 CORRECTIVE ACTION	5-17
7.0 WINCH DOCUMENTATION	5-20
8.0 SUMMARY	5-22
REFERENCES	5-23

1.0 INTRODUCTION

In the previous chapters a number of factors have been identified, which directly affect wire and cable life and have established criterion for the retirement of working cables. In order to maintain an operational control over the wire and cable in use it should be obvious that some system of documenting the life history of each individual wire or cable from purchase to retirement is necessary. In addition, winch performance, maintenance, and repair records are required in order to field and maintain a reliable and complete system.

The accurate recording of the "Life History" of individual wires and cables serves a number of important functions when conscientiously applied across the community. Primarily it acts as a management control tool at the institutional level, a quality control device from the standpoint of the scientific user, a valuable source of background information in the event of a wire failure, and as justification for replacement of a wire or cable when required. Simple attention to the details involved in recording required information can make this documentation process an easy, yet effective means of control. The objective of the documentation process is to provide the winch and wire system user with a demonstrable performance record and previous cable history.

From a carefully practiced procedure of documentation, it is possible for the vessel operator to assess his rope and cable inventory and to make accurate assessments of his future requirements, as well as presenting, upon request, a complete history of any wire in his inventory. This latter aspect is of importance where inter-institutional loans of wires occur. In this case, the borrower can receive a wire of known condition via its life history record.

The necessity for establishing a consistant means of documenting winch, wire and cable histories cannot be stressed too heavily. In a climate of rising product and instrument costs, each loss, due to the failure of a wire or cable, through unrecorded incidents, inadequate practices, or operation of a wire past its realistic retirement condition, directly affects the entire community. Since it is the task of the vessel operator to provide the scientist user with a reliable winch and wire system at sea, it is important that the operator know the condition of the wires in his inventory with a fair degree of reliability. This condition is achievable through the documentation process.

The idea in winch and wire documentation would be the use of a simple standard series forms within the oceanographic community. Such a standardized format would allow easy evaluation of borrowed wires, provide an engineerig data base previously unavailable, and provide important feedback to wire manufacturers for product improvement. The implementation of a

standard documentation format would in no way infringe on an institution's individuality, but would, instead, demonstrate the high level of cooperative effort that has been a part of this community.

2.0 WIRE AND CABLE DOCUMENTATION

Within this section we will concentrate on the development of a basic documentation scheme designed to encompass all of the aspects of a wire's useful life. The key points, which will be discussed, include wire identification, specification, use records, lubrication, wire termination, retirement, and where necessary, wire failure reporting. In addition, sample formats of suggested forms have been provided as a guide to establishing a comprehensive wire life history record.

As part of the documentation process, it must be realized that a strong centralized managerial structure is required to act as a repository for the wire and cable data generated in the field. Since it is impractical to expect any single individual or office to generate all of the information required of a comprehensive recording scheme, a high degree of reliance will have to be placed on the ships and technical support personnel. Once a clear division of labor is established, the central managerial control becomes responsible for only the maintenance of individual wire folders and input from the field programs.

2.1 Wire and Cable Identification

If accurate records are to be maintained, it will be necessary to establish a means of clearly identifying each individual reel of wire or cable within the inventory. Any simple alpha system can be adapted as long as it clearly identifies the individual wire or cable in some unique and non-repeatable form. The following illustrates one method of wire identification which can be employed.

FIGURE 5-1
WIRE IDENTIFICATION NUMBER

1	2	3	4	5	6
T	625	04	82	R	A

The particular alpha numeric combination shown above can be interpreted in the following manner:

Block 1 - This letter refers to the particular cable type, i.e., T = trawl wire, H = hydrographic, A - Acoustic or Electro-mechanical cable. The letter used only identifies the wire type as an aid in locating the reel in a storage area or during an inventory situation.

Block 2 - The three digit code shown in the example refers to the decimal diameter of the wire of that particular

FIGURE 5-1 (Cont'd)

reel. These digits would obviously change for different diameter wires. Since a wire's diameter can always be reduced to a decimal equivalent, the three digit code would form a simple means of further identification.

Blocks 3 and 4 - These two dual digit codes refer, respectively, to the month and year a specific reel of wire was received by an institution and became part of the inventory. The date at which a reel of wire is placed in service becomes important when high reliability of the wire is required. By including the date of receipt in the identification number, a ready reference is provided at a glace.

Block 5 - The single letter code in this block can be used to identify the institutional point of origin of a particular reel of wire or cable and can become important as wires are loaned or traded amongst instituitons. By being able to identify the institution managing the reel in use, its background and use record are always available.

Block 6 - This block would be reserved for the identification of simultaneous purchases of the same wire by an institution. In this case, either an A or a B suffix would be added to the code to differentiate the individual reels, thereby preserving their individuality.

The example described above is merely a sample of reel identification and there are certainly alternative approaches to such a system. However, the key to the success of an identification system is its simplicity and ease of use. While simplicity is the key, the need to preserve the individuality of each reel of wire or cable is paramount in the management scheme.

2.2 Wire Specifications

This information will be readily available from the original purchase order; a copy of which should be placed in the reel folder. The original ordering information, which makes up the wire specification, should include such items as original wire length, type of wire coating, material, lubrication, lay of the cable, etc. Given the ordering specifications provided in sections 1 and 2, it can be seen that the specifics of each reel of wire are important due to the options available.

2.3 Wire Documentation System

The following pages will describe the use and usefulness of the system illustrated in this chapter. It is certainly

WIRE USE RECORD

REEL NO. ____________

CRUISE NO. ____________ DATE ____________

LOWERING NO. ____________ STATION TYPE ____________

PAYLOAD WT. (AIR) ________ SEA STATE ____________

INSTALLED WIRE LENGTH ________________

AVAILABLE WIRE LENGTH ________________

TERMINATION TYPE ____________________________

MAXIMUM WIRE PAID OUT __________________ METERS

MAXIMUM TENSION OBSERVED ________________ LBS.

WIRE OUT METERS	OBSERVED TENSION	WINCH SPEED		REMARKS
		OUT	IN	

FIGURE 5-2

WIRE USE RECORD

CONTINUATION SHEET

REEL NO. ______________ CRUISE NO. ______________

DATE ______________ LOWERING NO. ______________

WIRE OUT	OBSERVED	WINCH SPEED		REMARKS
METERS	TENSION	OUT	IN	

FIGURE 5-3

not the only approach to this problem, but its simplicity and diversity of personnel involvement would tend to guarantee its success in the field. If the system discussed in this chapter is used as described, it will institute a level of control over the wire used at sea which is both efficient and reliable. Although additional forms could be generated, and probably will be, it is suggested that they be as an addendum to the ones described here instead of replacements.

It is a well-proven fact that if the complexity of a form is raised too high, the return from that form diminishes. The forms developed for this chapter will provide the manager with a comprehensive data set without excessive work on his part and will allow him to maintain control over his inventory with relative ease.

3.0 WIRE USE RECORD

This particular aspect of the documentation system may well be the most important item in the system as it is the record of the working life of the cable. When properly completed by the ship or technical personnel, it will reflect length of wire deployed at each station, maximum stress placed on the wire, a record of termination and testing, a record of the wires lubrication in the field, and a record of reduction in total wire length due to kinks, corrosion, etc. The level of difficulty in completing a form of this nature is minimal due to the fact that it is filled out during a lowering when data is readily available.

The followig sample from (Figure 5-2 and 5-3) represents a typical use record which contains the minimum data required to ensure managerial control. As will be noticed in Figure 5-2 and 5-3, the form is flexible enough to accommodate a wide range of information pertinent to the wire's use.

The wire use record shown in Figures 2 and 3 is a highly versatile form which can be used to record not only a sure history of a particular station, but also acts as a wire deployment and lubrication record, and a record of wire shortening due to damage. The information contained in this form has been reduced to a minimum while still maintainig key points of interest that would be used in any after-cruise analysis. The following explanation of the requested data will point out the relative importance of this particular form.

<u>Reel Number</u>: The number would be derived from the shore support personnel at the time the wire was installed on the vessel. By incorporating it in the "Use Form," a clear trail is established for that particular reel.

<u>Cruise Number and Date</u>: These are superfluous except for the fact that the cruise designator is important since a

reel of wire may be loaned to another institution during its working life. In this case, identification of the cruise and the other institution's use of the wire becomes an important part of the reel's life history.

Lowering Number: Since every deployment of a wire or cable consumes valuable ship time, that deployment should be identified by a number unique to the cruise. A specific lowering number should be assigned to every sampling station, streaming of the wire, and streaming it for the purpose of lubrication.

Station Type: This item simply identifies the instrument being deployed for the purpose of the lowering. Since piston cores, dredges, or mid-water trawls all place unique stresses on the wire, a great deal of guesswork is eliminated in later analysis of the station. Type is identified on the form. In addition, this item is also used to identify wire tensioning deployment as well as the wire lubrication process.

Paylod Weight (Air): Since the weight of any instrument attached to a reel of wire or cable should be known or can be determined easily, it is important that it be recorded as part of the permanent record. With a known instrument weight recorded for each station, both an on-the-spot or later analysis of wire performance can be easily achieved and wire status evaluated.

Installed Wire Length: This number refers to the length of wire which was installed on the winch at the time of vessel departure from port. The installed wire length will remain constant for the duration of a cruise. Since the potential cutting of small lengths of wire during the cruise is always present, this figure forms the basis for later evaluation of a remaining wire length when the vesesl returns to port.

Available Wire Length: This figure represents an available wire length for the specific lowering being conducted and the figure is adjusted based on sections which are cut off due to damage. For example, Figures 2 and 3, an available wire length of 10,480m is shown at the beginning of the lowering. At the end of the lowering (Figure 3), 20 meters of cable was removed due to damage. The available wire length for the next station has now been reduced to 10,460m and should be recorded as such on the next form.

Sea State/Weather: Since the motion of the vessel, due to sea conditions and weather, imparts stresses to the wire, their recording of these conditions becomes a necessary part of any later wire analysis and should be included as part of the permanent record.

Termination Type, Application, and Testing: In order to reduce the number of forms in this pilot system and to make

termination data readily available, it has been included in each sample use form. These sections are filled in based on the previous Use Form unless it has been necesary to terminate the wire. In this case, the new termination data is recorded and remains the same until another termination process is performed.

The recording of the test load applied to seat the fitting (in this case an Electroline Termination) is important to ensure reliability. The recording of the size of the fitting is important to ensure that the proper fitting was used.

Maximum Wire Out Maximum Tension: These two numbers are derived from the actual lowering record and are repeated at the top of the Use Form as a means of ready reference for wire analysis or the determination of a maximum wire length to be paid out during the lubrication process.

The bulk of the remainder of the form (illustrated in Figures 2 and 3), is self-explanatory. The frequency of recording the amount of wire over the side is an individual choice, but should be no less than once every 500 meters, or at the time of specific events, i.e., winch speed changes, corer pullout, etc. Major events, such as cutting sections of the wire, lubrication, winch malfunctions, and new terminations, should also be recorded as they occur pertinent to a specific lowering. The object of the sample form is to provide the maximum data with the least confusion and effort.

4.0 WIRE INVENTORY RECORD

This portion of the documentation system is designed to provide the user with a rapid assessment of his organization's wire inventory. It is keyed, as are all other forms in this system, on the wire identification number that has been established for each reel of wire. The Wire Inventory Form (Figure 5-4), requires very little effort on the part of the user to maintain while at the same time providing a comprehensive review of the organization's available ropes and cables.

The five (5) categories of information required by this form are explained as follows:

Reel Number: This number, as was explained in Section 2.1, is a unique identification of each reel of wire or electromechanical cable residing within the inventory. It is used here as a means of identification and cross-reference within the documentation system being explained.

Purchase Date and Length: The information called for here is easily derived from the original purchase order specifications and receiving report. Its inclusion in this form creates a starting point for the wire's use history as well as establishing an age for each wire in use. Over time, patterns

WIRE INVENTORY RECORD

REEL NO	PURCHASE		RETIREMENT		WIRE STATUS	REMARKS
	DATE	LENGTH	DATE	LENGTH		

FIGURE 5-4

will develop within each organization that will determine the useful life of each type of wire or cable being used from which predictions of future needs can be generated.

Retirement Dates and Length: When a wire is retired from service, that date should be entered on the inventory record along with the length of the wire at that time. Since this point represents the end of the wire's useful life as a deep sea tool, its period of service can be accurately calculated and use pattern determined. Retirement should be determined based on the recommendations laid down in chapters 1 and 2. If these recomendations are applied evenly across the community, an accurate picture of wire needs can be established over time.

Wire Status: Effectively this block of information details the final disposition of a reel of wire. Either it is scrapped upon retirement or maintained in storage for limited use in shallower waters. This last disposition can be important when it is realized that although the bulk of the wire may meet the retirement criterion, a certain portion of the wire, i.e., closest to the winch drum, will be essentially unused. Based on the wire use record, the available length can be determined and could potentially be used in support of smaller coastal research vessels which do not require a true deep ocean capability.

Remarks: This section can be used to record additional information pertinent to the wire during its use or after retirement, i.e., loaned to another organization, wire failures, etc.

5.0 SHORE BASE WIRE LOG

The third aspect of this documentation system concerns the handling of at-sea use data, storage, transfers, and routine testing of the operational wire within the inventory. The Shore Base Wire Log (Figure 5-5 and 5-6), is designed to incorporate the high points in an individual wire's operational life as well as presenting its status at any point in time. As in all forms conceived for this particular system, the Shore Base Wire Log is keyed to a specific reel number.

A copy of the wire log should be established for each reel of wire within the inventory with continuation sheets added to the file as they become necessary. The various blocks requiring input from the user are explained as follows.

Reel Number: The same as in all previous forms.

Date Received: This date should correspond to the date at which the reel of wire was received by the organization and became part of the inventory. This date can be derived from either the original recover report, the Wire Inventory Record, or the reel number itself.

<u>Manufacturer</u>: Self-explanatory.

<u>Wire Construction</u>: This line should contain a detailed description of wire size, construction (i.e., 3 x 19, 3 x 46, 6 x 19, etc.), or for electro-mechanical cables the armoring, number of conductors, etc.

<u>Location</u>: This block is used to explain the physical location of a reel of wire within the inventory so that its path can be traced throughout its useful life. When properly recorded, a wire travels from storage to sea or out on loan to another user can be accurately traced and use periods determined.

When in storage, the type of storage should be indicated such as inside or outside. While at sea, the particular cruise using the wire should be recorded, along with a vessel identification. During loans the organization receiving the wire should be listed.

<u>Dates</u>: This block should reflect the actual use period for each event shown in the location column.

<u>Testing</u>: A periodic test plan for each wire sould be established and the results of these tests reflected in the "Remarks" column of the form. Both mechanical tests (i.e., breakig strength), and electrical tests (continuity and resistivity of conductors), should be recorded as tests are performed.

<u>Lubrication</u>: This block should reflect the use of wire lubricants during the wire's various use periods. This information is derived from the Wire Use Record and its recording here only indicates that a lubrication process has occurred and during which period.

<u>Remarks</u>: This section of the form can be used to briefly describe significant events such as test results, high wire loading conditions, etc. The Remarks section merely acts as a flag to alert the user that an event of importance has occurred. Details of the event can usually be found in the Wire Use Record, if further investigation is required.

6.0 WIRE FAILURE REPORT

Although no one likes to consider wire failures, it is, more or less, important that when they occur they be fully and accurately reported. The minimization of a catastrophic failure by either refusing to face the problem or by simple acceptance of such a failure as a natural occurrence does nothing to eliminate the initial causes of the problem. By minimizing a loss all that is accomplished is the recreation of the same set of conditions that led to the failure in the first place and the

SHORE BASE WIRE LOG

REEL NO. ______________

DATE REC. ______________ MANUFACTURER ______________

WIRE CONSTRUCTION ______________

LOCATION (CRUISE - STO. - LOAN)	DATES		TESTING		LUBRICATE		AVAILABLE LENGTH (M)	REMARKS
	FROM	TO	MECH	ELEC	YES	NO		

FIGURE 5-5

SHORE BASE WIRE LOG
CONT. SHEET

REEL NO.__________

LOCATION	DATES		TESTING		LUBRICATE		AVAILABLE	REMARKS
(CRUISE - STO - LOAN)	FROM	TO	MECH	ELEC	YES	NO	LENGTH ('M)	

FIGURE 5-6

probability of another failure in the future is more or less assured.

There has been a tendency in the past not to discuss wire failures or to share with other users the circumstances surrounding a failure. This philosophy is both self-defeating and counter-productive to good wire management since it allows managerial conditions to perpetuate within the community. The efficiency of our wire use in the future would be much better served if information and an analysis of each catestrophic failure were freely circulated around the user community.

In order to derive a comprehensive record from a wire failure, the form shown in Figure 5-7 has been developed. As will be seen, it is comprised of three (3) separate sections dealing with the initial failure, a later engineering analysis and finally a description of any corrective action taken to correct problems identified during the engineering analysis. It is felt that this style of report should be circulated to other users engaged in the same type of at-sea work as a warning and as a preventative against similar failures. It should be remembered that unless adequate and accurate data is made available to users and manufacturers alike, it is not possible to influence product improvement or increase our at-sea reliability.

The Wire Failure Report (Figures 5-7 and 5-8), is designed to handle all data pertinent to the initial failure and can be augmented by witness' statements as appropriate or if desired. When fully completed, this report plus substantiating appendicies should provide a complete synopsis of the failure as it occcurred and any action taken to prevent a future recurrence.

6.1 SHIPBOARD ANALYSIS (FIGURE 5-7)

This section of the report should be completed at sea immediately following a catastrophic wire failure. The information requested at the top of the form is, essentially self-explanatory in nature, and can be readily completed. Beginning with the data concerning damage to the vessel, the information requested becomes a little less clear.

Damage to Vessel: This item is of major importance since damage to the vessel structure or sheave train may actually be a cause of the failure rather than the result of it. In this light all damaged parts should be recorded and, where possible, the broken pieces preserved for later analysis. All damaged items should be briefly listed in the space provided in order to begin to establish a sequence of events.

Equipment Lost: A brief description of all items of equipment lost should be listed in the space provided. The equipment may not be a direct cause of the failure, but its

WIRE FAILURE REPORT

SECTION I

SHIPBOARD ANALYSIS

REEL NO. ________ REPORT NO. ________

DATE OF FAILURE ________ CRUISE NO. ________

LOWERING NO. ________ STATION TYPE ________

SEA STATE ________ PAYLOAD WT ________ (LB-AIR)

WIRE TENSION AT FAILURE ________ (LBS)

WIRE OUT AT FAILURE ________ (M)

TOTAL WIRE LOST ________ (M)

DAMAGE TO VESSEL ________

EQUIPMENT LOST ________

LOCATION OF FAILURE ON SHIP ________

CAUSE OF FAILURE (ON SITE EVALUATION) ________

PREPARED BY ________

TITLE ________ DATE ________

FIGURE 5-7

description will provide useful data during a later analysis.

Location of Failure on Ship: It is crucial that the actual point of failure be identified as accurately as possible to aid the engineering analysis. Major differences in failure modes can be determined if the location of the break is known. For instance, a failure at the winch probably does not have the same cause as one occurring at the outboard sheave. The observations of eye-witnesses are an important ingredient in answering this question.

Cause of Failure (on-site evaluation): Once all pertinent data has been assembled, a rough cause of the falure should be determined and recorded in the space provided. This evaluation is, of course, preliminary, but is onetheless valuable in an assessment of the actual problem. Once this on-site evaluation is completed, it should be endorsed by either the vessel's captain or other responsible party.

6.2 ENGINEERING ANALYSIS (FIGURE 5-8)

This sheet has been developed to accompany the shipboard analysis of a wire failure at sea. Once the preliminary information concerning the failure has been received, the entire incident should be evaluated by qualified engineering personnel to determine the accuracy of the initial assessment and to recommend any corrective measures which may be required. A summary of the engineering analysis should be recorded on the form shown in Figure 5-8 and filed, with the initial report. Differences of opinion as to the cause of the failure should be documented and attached to this form as part of the permanent record.

The form itself is essentially self-explanatory and the only unique input concerns the number of appendicies attached to this section. This will vary from no appendidies when the analysis indicates that the preliminary work is accurate, to several appendicies at times of disagreement with the initial findings. As in the previous form, the author of the analysis should be identified at the bottom of the sheet.

6.3 CORRECTIVE ACTION (FIGURE 5-9)

Once the cause of a wire failure has been established, it is obviously necessary to institute some form of corrective action to ensure that a repeat failure will not occur. When a course of action has been selected, a summary of that action should be recorded on this form and added to the Wire Failure Report to complete the package. Corrective action may take the form of either procedural changes in its simplest form or may result in major structural or component changes in the winch system.

WIRE FAILURE REPORT

SECTION 2

ENGINEERING ANALYSIS (ON SHORE)

REEL NO.________________ REPORT NO.________________

APPENDICIES INCLUDED__________

SUMMARY OF ANALYSIS________________________________

PREPARED BY________________________________

TITLE________________________ DATE__________

FIGURE 5-8

WIRE FAILURE REPORT

SECTION 3

CORRECTIVE ACTION TAKEN

REEL NO.__________ REPORT NO.__________

PREPARED BY__________

TITLE__________ DATE__________

FIGURE 5-9

In any event, the Wire Failure Report in its entirety is designed to accomplish three (3) things: 1) document the failure, 2) analyze the failure, and 3) to produce recommendations for eliminating the cause of the initial failure. Although the wire failure reporting scheme presented here may appear lengthy and complicated, it should be remembered that wire failures are infrequent, but that when they do occur, they require careful analysis if a recurrence is to be prevented.

7.0 WINCH DOCUMENTATION

Given the variety of winches found on most research vessels and the fact that they are the primary working tool of the oceanographer, it is fairly obvious that they deserve special attention to ensure their operational readiness. This special attention should take the form of programmed maintenance, and repair combined with a procedure of reporting the performance of such maintenance. Traditionally records of winch maintenance have been buried in the ship's engineering log where they are difficult to extract on short notice, and the frequency of winch related work difficult to control.

A simple approach to this problem is the breaking out of winch related work in a specialized report or winch maintenance log. Since it deals only with winch related problems, a rapid assessment of maintenance schedules, etc., can be easily derived. This type of approach has been used successfully by the University of Rhode Island over the past two years with a noted increase in winch reliability.

The Winch Maintenance Log (Figure 5-10), was developed to monitor the condition and maintenance performed on each winch by the ship's personnel. In addition to routine care given the winches, the log should also reflect problems as they arise during operation. In this way a clear picture of each winch in use is maintained and recurring problems quickly identified.

The Winch Maintenance Log shown in Figure 5-10 is designed for use by the ship's engineering personnel with each winch system having its own separate log. All that is required in maintaining the log is that the vessel, winch type, i.e., trawl, hydrographic or acoustic, be identified and a corresponding serial number be entered at the top of the page. Dates of maintenance and repair should be entered as they occur as well as the initials of the individual performing the work.

Description of Maintenance or Repair: This section of the log should clearly describe the type of work that has been performed as well as identifying any parts that are replaced or repaired. Typical items which would be listed here include greasing of the winch, replacement of hydraulic hoses and their inspection, major component replacemet, etc. In effect this form becomes the life history record of the winch and when

WINCH MAINTENANCE LOG

VESSEL ____________

WINCH TYPE ____________ SERIAL NO. ____________

DATE OF WORK	BY	DESCRIPTION OF MAINTENANCE OR REPAIR

FIGURE 5-10

carefully completed can be used to identify recurring problems which affect winch efficiency and reliability.

8.0 SUMMARY

The documentation system that has been described in the preceding sections constitute a basic minimum that would be required to effect control over both an organizational wire inventory and associated winch systems.

The success of this or any other documentation scheme rests with the individuals who are responsible for submitting the required information and their cooperation can be more easily achieved if the demands on their time are kept to a minimum. In other words, the proliferation of unique forms and reports should be kept to a minimum if the system is to have any real value.

It is strongly encouraged that the documentation scheme, no matter what it is, be established as a standard within an organizational group such as UNOLS, NOAA, etc. Once this is accomplished, the control over and reliability of the wires and winches within these organizations becomes a realistic approach to the problem.

REFERENCES

A Study of CTD Cable and Lowering Systems. H.O. Berteaux et al. WHOI-79-81. Report on Ship Procedures - URI.

CHAPTER 6

Rope and Cable Terminations

C. ROBERT SHAW
AND
W. A. LUCHT

1.0 INTRODUCTION 6—2

2.0 SELECTION CRITERION 6—3

2.1 TYPE AND SIZE OF WIRE OR CABLE 6—3

2.2 CORROSION POTENTIAL 6—3

2.3 LOADING AND CABLE FATIGUE 6—4

2.4 TERMINAL EFFICIENCY 6—4

2.5 ASSEMBLY REQUIREMENTS AND COST 6—4

3.0 INSTALLATION PROCEDURES 6—5

3.1 WIRE ROPE CLIPS 6—5

3.2 WIRE ROPE THIMBLES 6—9

3.3 OPEN WEDGE TERMINATIONS 6—10

3.4 POURED ON SPELTER SOCKETS 6—11

3.5 COMPRESSED SLEEVES (NICOPRESS) 6—19

3.6 SWAGED TERMINATIONS 6—21

3.7 MECHANICAL TERMINATIONS (ELECTROLINE OR FIEGE) 6—22

4.0 ELECTRO-MECHANICAL TERMINATIONS 6—28

4.1 ELECTROLINE E-M CABLE TERMINATIONS 6—28

4.2 INSTALLATION PROCEDURES 6—29

4.3 COMBINATION MECHANICAL/EPOXY TERMINATIONS 6—33

4.4 HELICALLY WOUND TERMINATIONS 6—37

1.0 INTRODUCTION

Selection of the proper type of wire or cable termination used in oceanographic applications is a key factor in the safe and effective utilization of the winch and wire system in the deep sea. The purpose of this section will describe the seven basic types of wire rope terminations, as well as those used with electro-mechanical cable. In particular, the characteristics, advantages, limitations, and assembly procedures for each type will be fully illustrated.

The selection of a wire or cable termination for a particular application should be considered carefully, bearing in mind that there are advantages and disadvantages to each termination type discussed in this section. To select the correct type of terminal for a particular application, the following factors should be carefully evaluated:

- Type and size of cable involved
- Corrosion potential
- Loading and cable fatigue
- Efficiency required
- Assembly requirements and cost

Primarily, seven basic types of wire rope terminations will be discussed as follows:

a. Wire rope clips - A U-bolt and saddle combination or a "fist-grip" nut and bolt arrangement used to fasten a loop of wire rope that is formed around a thimble.

b. Open-wedge terminations - Also called "wedge sockets"; the cable is looped around a wedge which is inserted into a socket or "basket" and held secure by tension on the line.

c. Poured-socket terminations - Also known as Spelter sockets; molten zinc or an epoxy compound is poured into the socket to bond the wire rope inside the fitting.

d. Compressed sleeves - Also known as Nicopress terminations; sleeves are crimped or compressed around the cable, usually by use of handtools.

e. Swaged terminations - Attached by cold forming under high pressure so that the metal of the fitting flows around and between strands and wires of the rope.

f. Mechanical terminations - Also known as Electroline fittings; these devices utilize wedge or plugs of various sizes and configurations to hold the cable inside a threaded lock sleeve.

g. Helical terminations - Also known as Preformed Dyna-grip terminations. This device utilizes helical gripping

wires which wrap around the cable and are finished in a thimble or epoxy filled fitting.

2.0 SELECTION CRITERION

As mentioned in the introduction, a series of five criterion should be considered in the selection of a wire or cable termination. The following discussion of these criterion will provide insight into the problems which can be encountered in the selection process.

2.1 Type and Size of Wire or Cable

The terminations selected must be compatible with the type of cable being used and must result in the maximum effective holding strength. For example, swaged terminations, compressed sleeves and wire rope clips are not efficient terminals for hemp-core wire rope, armored electrical cables or synthetic cables. Application of such terminations requires squeezing them on to the cable, and "soft-core" cable material will give way under the pressure, thereby weakening the effectiveness of the termination. Mechanical, poured-socket and open-wedge terminations can be use effectively with these types of cables since they achieve their efficiency from bonding or compressing only the steel of the wire or cable.

Cable size is a major consideration because of the standard capacities of termination devices that are generally available. Compressed sleeves can be used on cables up to 5/8 inch diameter. Wire rope clips, open wedges and mechanical fittings are standard for cables up to 1 1/2 inches in diameter. Swaged terminals can be obtained as large as 2 1/2 inches and poured sockets up to 4 inches.

2.2 Corrosion Potential

The corrosion problems experienced in an oceanographic application, when wires and terminations are subjected to alternate immersion and drying cycles, is well known. Given this as an existing condition, it is important to consider the standard materials in which the various termination devices are available. In the main, poured sockets, helical terminations, wire rope clips, and open-wedge sockets are stocked only in steel, although a zinc plating of these terminations is available. Compressed sleeves (Nicopress), swaged terminations, and mechanical fittings (Electroline), are available in a wide variety of materials ranging from steel to stainless steel. Compressed sleeves and the mechanical fittings are also produced, in certain sizes, in both bronze and aluminum for special applications. The variety of materials available in termination construction make the matching of a specific fitting to a particular application a fairly simple process.

2.3 Loading and Cable Fatigue

All seven basic types of terminations are suitable for static and moderately cyclic loads such as those imposed by cranes, hoists, guy wires, tie downs, slings, etc. Only the mechanical and helical terminations, however, are designed to accommodate the shock and vibration loads imposed by winches, buoys and towed bodies in the marine environment.

The mechanical fittings have a "transition zone" in the nose where the cable enters the termination. In this "semi-loose" transition zone, the tension, compression and bending stresses in the rope strands are dissipated. Because of the ways in which other types of terminations are affixed to the cable, they have a hard transition from the cable to the terminal which can contribute to shorter cable fatigue life. The helical type termination provides a long cable life by dissipating the shock and vibration loads in the spring action of the helical gripping wires.

2.4 Terminal Efficiency

The more efficient the terminal, the smaller and lower cost the cable may be. This also can affect the cost of winches and other handling equipment.

The swaged and helical terminals are rated for 100% of the cable's rated braking strength. Poured sockets, compressed sleeves and mechanical fittings are rated at 95% to 100%, while wire rope clips and open-wedge terminations are rated at 75% to 85%. Wire rope clips tend to lose their holding strength with use and must be retightened from time-to-time. At the other end of the scale are the mechanical and open-wedge terminals in which the wedging action actually increases efficiency with loadings (Table 1).

2.5 Assembly Requirements and Cost

Wire rope clips are both the least expensive and easiest termination that can be applied in the field. They require only attention to clip spacing, placement, and tightening torque to perform efficiently. The open-wedge termination is only slightly more expensive, but is just as simple to install, requiring only hand tools for the application. Although the simplest, they are also the least efficient of the six terminations discussed. The compresseed sleeve (Nicopress) fitting represents another low cost, but highly efficient means of terminating a wire rope. Special tooling, which is available from the manufacturers, is available to ensure proper compression of the sleeve and the installer needs only to match the required number of compression to the sleeve size selected for use. Used in the proper situation, the compressed sleeve can be rapidly and efficiently reapplied in the field with no special training.

The swaged and poured socket (Spelter) terminations represent a moderately priced fitting with a high efficiency rating. Swaged terminations require large hydraulic presses for proper installation and do not readily lend themselves to reapplication in the field under most circumstances. The poured socket represents a highly efficient fitting which can be reapplied in the field using either a molten zinc or an epoxy resin to achieve wire bonding. This approach, however, requires careful attention to detail and the use of an aid to clean the wire ends prior to fitting installation.

The mechanical termination represents a more expensive fitting type discussed in this section due to the number of components involved in each assembly. Although it appears to be a complicated termination, it can be easily installed without special equipment and with only the training received from the manufacturers' literature. Attention to assembly detail and adequate proof loading are all that is required to produce a highly efficient termination using this fitting.

The helical terminal is the most expensive. Assembly can be accomplished easily in the field with no special training or equipment. However there is a 24 hour cure for the epoxy filling.

Inspection is another important assembly consideration. The swaged and compressed sleeve terminals can be inspected for effective assembly by measuring the final diameters. Wire rope clips can be inspected with a torque wrench. The poured socket cannot be inspected to determine if the assembly is proper. The mechanical terminal has an inspection hole built in to facilitate visual checking. The helical termination can be inspected to assure that no wires are crossing themselves and that the body is filled with epoxy.

Perhaps the single most important thing to remember in the selection of a termination for either wire rope or cable is that the fitting should be chosen at the same time as the cable is specified. A second major consideration is physical size of the termination relative to the instrument or device it will be attached to. This is especially important when the fitting is required to pass through an instrument as in the case of a piston coring device or over a sheave. In these cases physical size and configuration of a fitting will influence the type selected. Since the wire or cable termination is vital to the safe and efficient use of the wire or cable, it should be viewed as an integral part of the system and gives careful consideration when purchased.

In order to select the proper terminations for an application, the factors discussed here should be carefully considered as well as those presented in Table 1.

CHARACTERISTICS OF SEVEN TYPES OF TERMINALS FOR WIRE ROPE AND CABLE

TYPE OF TERMINAL	TYPE OF ROPE/CABLE	ROPE SIZE	STANDARD MATERIALS	DESIGN LOADING	EFFICIENCY % OF RBS	CABLE LIFE	ASSEMBLY	COST
Wire Rope Clips	IWRC Strand	3/16 - 1 1/2	Steel	Static Cyclic (needs re-adjustment)	75% - 85%	Short	No training needed. No special tools. Fast field assy and disassembly, Reuseable. Torque inspection.	Low
Open Wedge	IWRC Hemp Center Strand Synthetic	3/8 - 1 5/8	Steel	Static Cyclic (effectiveness increases with load)	75% - 85%	Short	No training needed. No special tools. Fast field assy and disassembly, Reuseable. Visual Inspection.	Medium
Poured Socket	IWRC Hemp Center Strand	3/16 - 4	Steel	Static Cyclic	95% - 100%	Long	Training required. Special tools and equipment req'd. Difficult field Assy. Not reuseable. No visual inspection of acceptable assy.	Medium
Compressed Sleeve (Nicopress)	IWRC Strand	1/16 - 5/8	Steel Stainless Aluminum Bronze	Static Cyclic	95% - 100%	Short	No training req'd. Special tools req'd. Field assy on only smaller sizes. Not reuseable. Dimensional Inspection.	Low
Swaged Socket	IWRC Strand	1/8 - 2 1/2	Steel Stainless	Static Cyclic	100%	Medium	Training needed. Tools and Equipment req'd. No field assy. Not reuseable. Dimensional Inspection	Medium
Mechanical (Electroline Flege)	IWRC Strand Hemp Center Double-armored Synthetic	1/16 - 1 1/2	Steel Stainless Aluminum Bronze	Static Cyclic Shock Vibration (effectiveness increases w/load)	95% - 100%	Long	No training needed. No special tools. Fast field assy, reuseable. Visual inspection.	High
Mechanical (Preformed Dyna-Grip)	IWRC Hemp Center Strand Double-armored	1/8 - 3/4	Steel	Static Cyclic Shock Vibration	100%	Long	No training needed. No special tools. 24 hour cure for the epoxy. Not reuseable No method of inspection.	High

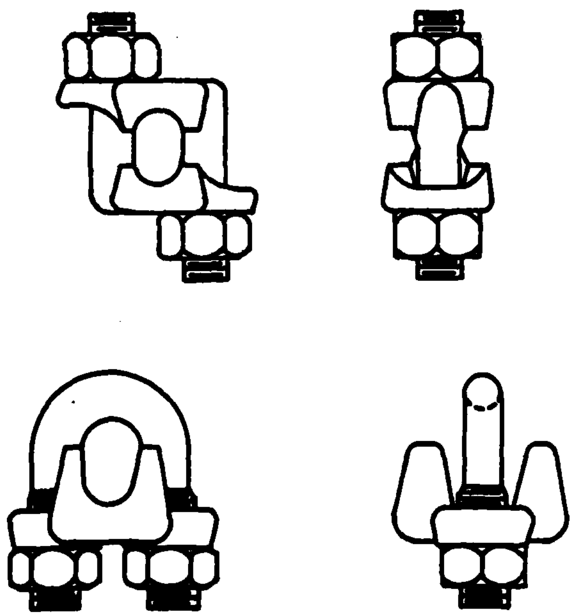
TABLE I

3.0 INSTALLATION PROCEDURES

In order to ensure the highest possible reliability in applying the particular termination selected for use, the following procedures have been detailed along with information which affects long-term performance. For any type of termination there are "tricks of the trade" which assure the integrity of the fitting once applied, and which should not be deviated from if full reliability is to be achieved. The following procedures, if carefully followed, will assure the reliability required in an oceanographic application.

3.1 Wire Rope Clips

Wire rope clips (Figure 6-1), are sized and marked on the body of the clip with the wire diameter they are intended to be used with. It is important that the clip be matched to the diameter wire in use as mismatches will result in a drastic reduction in termination holding power and efficiency. Placement of the wire rope clip is of prime importance to achieve maximum holding power. It should be noted that most available wire rope clip data sheets specify only a minimum number of clips needed for ordinary loads. Where heavy loading is anticipated, it is strongly recommended that two additional clips be added to each installation.



WIRE ROPE CLIPS
FIGURE 6-1

The recommended procedure for applying wire rope clips to achieve the maximum termination holding power is as follows:

- Turn back the amount of wire required based on Table 2. This distance is measured from the thimble to the bitter end of the wire.

- The U-Bolt portion of the clip must be placed over the bitter end of the wire while the saddle of the clip is placed on the standing part of the wire. Any reversal of this procedure or a staggering of the clips will result in reduced efficiency of the termination.

TABLE 2

Wire Rope Clip Assembly Data **

ROPE DIA	NO. CLIPS REQ NORM	NO. CLIPS REQ H.D.	CENTER TO CENTER CLIP SPACING	LENGTH OF ROPE TURNBACK -INCH NORM	LENGTH OF ROPE TURNBACK -INCH H.D.	TIGHTENING TORQUE FT. LBS.
1/8	2	2	1 3/8"	3 1/4"	6"	5
3/16	2	4	1 7/16"	3 3/4"	6 5/8"	9
1/4	2	4	1 7/8"	4 3/4"	8 1/2"	18
5/16	2	4	1 7/8"	5 1/4"	9	30
3/8"	2	4	2 5/8"	6 1/2"	11 3/4"	42
7/16	2	4	2 7/16"	7"	11 7/8"	70
1/2	3	5	3 15/32"	11 1/2"	14 31/3"	75
9/16	3	5	3 1/2"	12"	19"	100
5/8	3	5	3 1/2"	12"	19"	100
3/4	4	6	4 1/4"	18"	27"	150
7/8	4	6	4 1/4"	19"	27 1/2"	240
1	5	7	4 3/4"	26"	35 1/2"	250
1 1/8	6	8	5 3/8"	34"	37 5/8"	310
1 1/4	6	8	5 3/4"	37"	48 1/2"	460
1 3/8	7	9	5 15/16"	44"	55 7/8"	520
1 1/2	7	9	6 1/2"	48"	60 7/8"	590
1 5/8	7	9	6 29/32"	51"	64 27/3	730
1 3/4	7	9	7"	53"	66 1/2"	980
2	8	10	8 15/32"	71"	87 31/32"	1340
2 1/4"	8	10	8 5/8"	73"	90 13/32"	1570
2 1/2	9	11	8 27/32"	84"	101 11/16"	1790
2 3/4	10	12	9 9/16"	100"	118 15/16"	2200
3	10	12	10"	106"	125 1/4"	3200

** Table based on Crosby Group Data

- The first clip should be installed within one saddle width of the end of the turned back wire and the nuts evenly tightened. The second clip should be installed as near the thimble or loop as is possible with nuts firmly installed, but not tightened.

- Space additional required clips evenly between the two clips already on the wire. Light tension should be applied between the terminal loop and the standing part of the cable before tightening all clips to their recommended torque. This process will eliminate slack occurring in the bitter end of the cable and produce a more uniform application.

- An initial load should be applied to the termination and all nuts retightened to their recommended torque prior to use of the termination. Once applied in accordance with the above procedures, the resulting termination will have an efficiency rating of 75-85% of the breaking strength of new wire.

- When a wire rope clip type of termination is used, it is advisable to periodically tighten the nuts to their recommended torque since wire vibration can result in a loosening of the U-bolt nuts.

3.2 <u>Wire Rope Thimbles</u>

The use of a thimble is highly recommended with this type of termination and some discussion of thimble styles is necessary at this point. Essentially, thimbles fall into three broad categories: 1) standard Wire Rope Thimbles, 2) Extra Heavy Thimbles, and 3) Solid Thimbles. Specific data relating to each style discussed will be found in the Useful Information section at the end of this handbook.

a. <u>Standard Wire Rope Thimbles</u> - This class of thimble is designed primarily for use in light duty situations where loading is minimal. Their use in heavy duty situations will result in a complete deformation of the thimble and the placing of excessive stress on the wire at the head of the loop. Under this situation, a failure of the wire can be expected, which will occur below the rated strength of both the termination and the wire.

b. <u>Extra Heavy Wire Rope Thimble</u> - This style of thimble has been designed for heavy duty service where high loading conditions are expected to occur. They are far more resistant to deformation due to loading and work to maintain an even wire loading condition in the loop of the termination. As

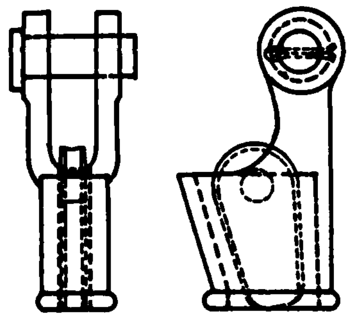
in all thimbles under load, the size of the pin used to attach the load to the cable is critical. Its diameter should be closely matched to the internal diameter of the thimble in order to reduce point loading.

c. Solid Thimble - This thimble is designed as the ultimate in crush-proof thimbles due to its solid steel construction. Primarily a unit for very heavy load conditions, it has a single disadvantage in that the hole sizes available are more limited than those found in the extra-heavy thimble. Where heavy loading situations are anticipated on a regular basis, it is recommended that a solid thimble be considered.

3.3 Open-Wedge Termination

Although this type of termination is typically found on crane cables, it has occasionally been used for limited trawl wire operations. Because of its diverse usage in the field, it is felt prudent to provide proper assembly instructions for this type of fitting.

The open wedge termination (Figure 6-2), although a simple style of fitting to install, should be approached with a certain amount of caution during installation of the wire. Improper placement of the wire can result in excessive wire stress at the termination resulting in a reduction of wire loading potential.



OPEN WEDGE TERMINATION
FIGURE 6-2

a. Installation Procedure - The simplicity and rapid installation potential of the open-wedge can be further enhanced by attention to a few details calculated to produce the maximum efficiency from this style of termination. The proper installation procdure is as follows:

- An inspection of both the socket and wedge should be made to identify any rough or burred surfaces on the wire path which could damage the wire under load. If irregularities are

discovered, they should be removed, if possible, or the socket or wedge replaced.

- The bitter end of the wire should be clean cut and served in order to prevent unlaying. It is important that the bitter end be clean cut rather than fused due to cutting with a torch in order to allow the individual wire strands to adjust around the sharp bend of the wedge. If the wire end is fused on installation, the movement of individual wire strands will be translated to the standing part of the wire causing irregularities in shape and unequal loading.

- To install the wire in the socket, the socket must be in an upright position (ears downward). The wire is then brought into the socket to form a large, easily handled loop. Care should be taken to ensure that the standing part of the wire is in line with the sockets ear (Figure 6-2).

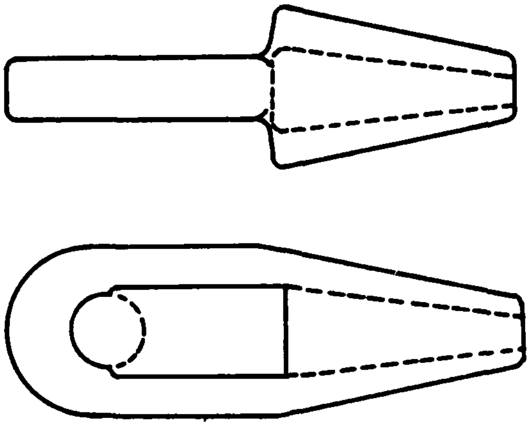
- The bitter end of the wire should extend above the socket for a distance equal to nine (9) times the diameter of the wire used. At this point the wedge is placed in the socket and a wire rope clip placed around the bitter end of the wire by clamping it around a short length of wire which has been attached to the bitter end to provide the mass required for the wire clips seating. The U-bolt of the wire clip should bear on the bitter end and the saddle on the added short piece of wire.

- By securing the fitting to a convenient pad eye or bitt, a load should be placed on the standing part of the wire. This load is steadily applied until the wedge and wire are pulled into position with enough strain to hold them in place when the load is released. During the seating of the wedge, sudden surge or shock loading should be avoided.

3.4 Poured or Spelter Sockets

The spelter socket (Figure 6-3) represents a highly reliable termination with a 100% efficiency when properly applied. The key factor in achieving the 100% efficiency of this termination is careful cleaning of the wire ends with a solvent solution and the positioning of the socket on the wire prior to the pouring of either the zinc or resin used to hold it in place. The cleaning of the wire ends allows for the maximum bonding action of the filler chosen while exact positioning of the

socket on the wire ensures an even loading of the wire strands in the field.



POURED SOCKET
FIGURE 6-3

A certain amount of caution should be used when dealing with this style of termination. The rigidity which is caused by the bonding action of the zinc or resin on the end of the wire causes a sudden dampening of wire vibration at the point where the wire enters the fitting. This dampening of vibration can lead to fatiguing of the wires at this point and frequent, careful inspection of the area for broken wire should be made. The detection of any broken wires should dictate the immediate replacement of the fitting.

a. Zinc Socketing Procedures (Figure 6-4) - The procedures for applying zinc-poured sockets is as follows:

- Measure the rope ends for socketing and apply serving at the base of the socket. As indicated in Figure 6-4a, the length of the rope end should be such that the ends of the wires when unlaid from the strands will be at the top of the socket basket. Apply a tight wire serving band for a length of two rope diameters beginning at the base of the socket and extending away from it.

- Broom out strands and wires in the strands. See Figure 6-4b and 6-4c. Unlay and straighten the individual strands of the rope and spread them evenly so they form an included angle of approximately 60°. If the rope has a fiber core, cut out and remove the core as close to the serving band as possible. Unlay the wires from each individual strand for the full length of the rope end, being careful not to disturb or change the lay of the wires and strands under the

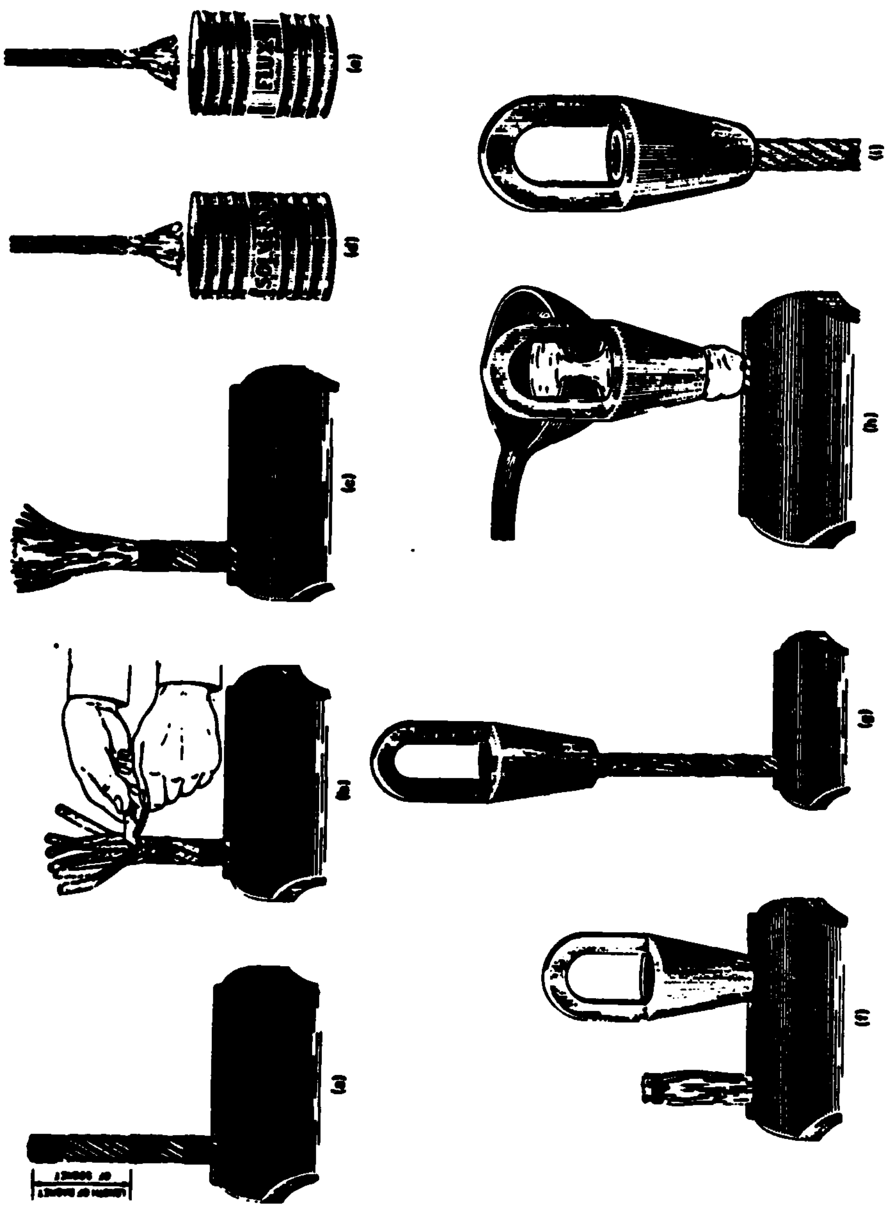


FIGURE 6-4

Attaching Wire Rope Sockets

serving band. If the rope has an independent wire rope core (IWRC), unlay the wires of the IWRC in the same manner.

- Clean the broomed-out ends. See Figure 6-4d. A suggested solvent for cleaning is SC-5 Methyl Chloroform. This solvent is also known under the names of Chlorothane VG or 1,1,1-trichloroethane.

  CAUTION: Breathing the vapor of chlorinated solvents is harmful; use only with adequate ventilation. Follow the solvent manufacturer's instructions; observe the label instructions.

  When using a solvent, swish the broomed-out rope end in the solvent and vigorously brush away all grease and dirt making sure to clean all the wires of the broomed-out portion to a point close to the serving band. A solution of hydrochloric (muriatic acid) may be used for additional cleaning. However, if acid is used, the broomed-out ends of the rope should be subsequently rinsed in a solution of bicarbonate of soda to neutralize any acid that may remain on the rope. Care should also be exercised that acid does not enter the core, particulary if the rope has a fiber core. Ultrasonic cleaning is a preferred method for cleaning rope ends for socketing.

  After cleaning, put the broomed-out ends upright in a vise until it is certain that all the solvent has evaporated and the wires are dry.

- Dip the broomed-out rope ends in flux. See Figure 6-4e. Make a hot solution of zinc-amonium chloride flux such as Zalcon K. Use a concentration of one pound of zinc-amonium chloride in one gallon of water and maintain the solution at a temperature of 180° to 200°F. Swish the broomed-out end in the flux solution, put the open end upright in the vise, and permit all wires to dry thoroughly.

- Close rope ends and install the socket. See Figure 6-4f and 6-4g. Use clean wire to compress the broomed-out rope end into a tight bundle so that the socket can be slipped over the wires. A socket should always be cleaned and heated before placing it in the rope. The heating is necessary to dispel any moisture and to prevent premature cooling of the zinc.

  CAUTION: Never heat a socket after it has been placed on the rope because of the hazard of heat damage to the wire rope.

When the socket has been put on the rope end, the wires should be evenly distributed in the socket basket so that zinc can surround every wire. Use utmost care to align the socket with the center line of the rope and to ensure that there is a vertical, straight length of rope exiting the socket that is equal to a minimum of 30 rope diameters. Seal the base of the socket with fire clay or putty, but be sure this material is not inserted into the base of the socket; if this were done, it would prevent the zinc from penetrating the full length of the socket basket and would create a void which would collect moisture when the socket is placed into service.

- Pour the zinc. See Figure 6-4h. Use zinc that meets the requirements in ANSI/ASTM B6-70, Specification for Zinc Metal (Slab Zinc), for "high grade" or Federal Specification QQ-Z-351-a Amendment 1, interim Amendment 2. Pour the zinc at a temperature of approximately 950°F to 975°F making allowances for cooling if the zinc pot is more than 25 feet from the socket.

  CAUTION: Do not heat zinc above 1100°F or its bonding properties will be lost.

  The temperature of the zinc may be measured with a portable phyrometer or a Tempilstik. Remove all dross before pouring. Pour the zinc in one continuous pour to the top of the socket basket so that all the wire ends are covered; there should be no "capping" of the socket.

- Remove the serving band. See Figure 6-4i. Remove the serving band from the base of the socket and check to see that zinc has penetrated to the base of the socket.

- Lubricate the rope. Apply a wire rope lubricant to the rope at the base of the socket and on any section of the rope from which the original lubricant has been removed.

b. Procedure for Thermoset Resin Socketing of Wire Rope

- General - Before proceeding with thermoset resin socketing, the manufacturer's instructions for using this product should be carefully read. Particular attention should be given to sockets that have been designed specifically for resin socketing.

- Seizing and Cutting the Rope - The rope manufacturer's directions for a particular size or construction of rope should be followed with regard to the number,

position length of seizings, and the seizing wire size to be used. The seizing which will be located at the base of the installed fitting must be positioned so that the ends of the wires to be embedded will be slightly below the level of the top of the fitting's basket. Cutting the rope can best be accomplished by using an abrasive wheel.

- Opening and Brooming the Rope End - Prior to opening the rope end, place a short temporary seizing directly above the seizing that represents the base of the broom. The temporary seizing is used to prevent brooming the wires the full length of the basket and also to prevent the loss of lay in the strands and rope outside the socket. Then move all seizings between the end of the rope and the temporary seizing. Unlay the strands comprising the rope. Open each strand of the rope and broom or unlay the individual wires.

  When the brooming is completed, the wire should be distributed evenly within a cone so that they form an included angle of approximately 60°. Some types of sockets require a different brooming procedure and the manufacturer's instructions should be followed.

- Cleaning the Wires and Fittings - Different types of resin with different characteristics require varying degrees of cleanliness. For some, the use of a soluble oil for cleaning wires has been found to be effective. For one type of polyester resin on which over 700 tensile tests on ropes in sizes 1/4 to 3-1/2 inches in diameter were made without experiencing any failure in the resin socket attachment, the cleaning procedure is as follows:

  Thorough cleaning of the wires is required to obtain resin adhesion. Ultrasonic cleaning in recommended solvents such as trichloreoethylylene or 1-1-1 trichloroethan or other non-flammable grease-cutting solvents is the preferred method of cleaning the wires in accordance with OSHA Standards. Where ultrasonic cleaning is not available, brush or dip-cleaning in trichloroethane may be used; but fresh solvent should be used for each rope and fitting and discarded after use. After cleaning, the broom should be dried with clean compressed air or in some other suitable fashion before proceeding to the next step. The use of acid to etch the wires before resin socketing is unnecssary and not recommended. Since there is a variation in the properties of different resins, the manufacturer's instructions should be carefully followed.

- Placement of the Fitting - Place the rope in a vertical position with the broom up. Close and compact the

broom to permit insertion of the broomed rope end into the base of the fitting. Slip on the fitting, remove any temporary banding or seizing as required. Make sure the broomed wires are uniformly spaced in the basket with the wire ends slightly below the top edge of the basket; make sure that the axis of the rope and the fittings are aligned. Seal the annular space between the base of the fitting and the existing rope to prevent leakage of the resin from the basket. A non-hardening butyl rubber-base sealant gives satisfactory performance. Make sure that the sealant does not enter the base of the socket so that the resin may fill the complete depth of the socket basket.

- Pouring the Resin - Controlled heat-curing (but without open flame) at a temperature range of 250° to 300°F is recommended -- and is essential if ambient temperatures are less than 60°F. When controlled heat curing is not available and ambient temperatures are not less than 60°F, the attachment should not be disturbed and tension should not be applied to the socketed assembly for at least 24 hours.

- Lubrication of Wire Rope after Socket Attachment - After the resin has cured, relubricate the wire rope at the base of the socket to replace the lubricant that was removed during the cleaning operation.

c. Description of the Resin - General -- Resins vary considerably according to the manufacturer; it is important to refer to the manufacturer's instructions before using resins as no general rules about them can be established.

Properly formulated thermoset resins are acceptable for socketing. These resin formulations, when mixed, form a pourable material that hardens at ambient temperatures or upon the application of moderate heat. No open-flame or molten-metal hazards exist with resin socketing since heat-curing, when necessary, can only be carried out at a relatively low temperature (250° to 300°F) that can be supplied by electric-resistance heating.

Tests have shown satisfactory wire rope socketing performance by resins having the properties of a liquid thermoset material that hardens after mixing with the correct proportion of catalyst or curing agents.

Properties of Liquid (Uncured) Material - Resin and catalyst are normally supplied in two separate containers, the complete contents of which, after thorough mixing, can be poured into the socket basket. Liquid resins and catalyst should have the following properties:

1) Viscosity of Resin-Catalyst Mixture - The viscosity of the resin-catalyst mixture should be 30,000 to 40,000 CPS at 75°F immediately after mixing. Viscosity will increase at lower ambient temperatures and resin may need warming prior to mixing in the catalyst if ambient temperatures drop below 40°F.

2) Flash Point - Both resin and catalyst should have a minimum flash point of 100°F.

3) Shelf Life - Mixed resin and catalyst should have a minimum of one year shelf life at 70°F.

4) Pot Life and Cure Time - After mixing, the resin-catalyst blend should be pourable for a minimum of eight minutes at 60°F and should harden in fifteen minutes. Heating of the resin in the socket to a maximum temperature of 250°F is permissible to obtain full cure.

Properties of Cured Resin

1) Socket Performance - Resin should exhibit sufficient bonding to solvent-washed wire in typical wire rope end fittings to develop the nominal strength of all types and grades of rope. No slippage of wire is permissible when testing resin-filled rope socket assemblies in tension; however, after testing, some "seating" of the resin cone may be apparent and is acceptable. Resin adhesion to wires shall also be capable of withstanding tensile shock loading.

2) Compressive Strength - The minimum compressive strength for fully cured resin should be 12,000 lb/in^2.

3) Shrinkage - Fully cured resin may shrink a maximum of 2%. The use of an inert filler in the resin is permissible to control shrinkage if the viscosity provisions specified for the liquid resin are met.

4) Hardness - A desired hardness of the resin is in the range of Barcol 40-55.

Resin Socketing Compositions - Manufacturer's directions should be followed in handling, mixing, and pouring the resin composition.

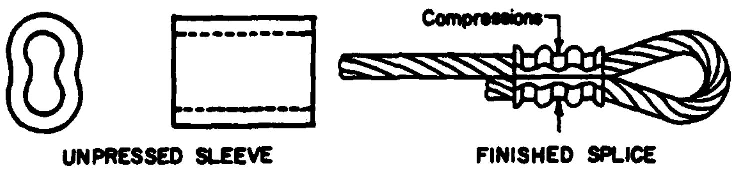
Performance of Cured Resin Sockets - Poured resin sockets may be moved when the resin is hardened. After the ambient or elevated temperature cure recommended by the manufacturer, resin sockets should develop the nominal strength of the rope, and should also withstand shock loading sufficient to break the rope without cracking or breakage. Resin socketing materials that have not been tested to these criteria by the manufacturer should not be used.

3.5 Compressed Sleeves (NICOPRESS)

The compressed or Nicopress sleeve (Figure 6-5) represents a style of wire and cable termination which has been available for the past thirty yeras. Its high efficiency, 95% - 100% of the breaking strength of the wire, and its simplicity of installation have made it an ideal type of termination for general field use. The success of the compressed sleeve is dependent upon the selection of the proper sleeve to match the wire to be terminated and matching the compression requirements of the sleeve. Specialized tooling is produced which will ensure proper compression to achieve the maximum holding power of the fitting.

When terminating a rope or cable, both the sleeve and tool should match the requirements of the cable size. Table 3 provides a guide to this selection as well as the recommended compressions needed for maximum efficiency.

It should be noted that in Table 3 the number of sleeves, required per instalation, to achieve the maximum holding power is the same for all wire sizes. The manufacturer recommends a single sleeve per termination and the addition of more sleeves in no way increases the ultimate holding power of this type of termination. One factor which can affect the efficiency of the compressed sleeve is excessive compression. The recommended compressions shown in the table and in Figure 6-5, allow for ultimate holding while providing an adequate stress relief at both ends of the sleeve. This factor can become extremely important when stainless steel sleeves are used in the termination of a wire.



COMPRESSED SLEEVE (NICOPRESS)
FIGURE 6-5

As with other types of terminations which require the wire to be looped at the termination point, it is necessary to use a properly sized thimble to protect the wire. The discussion found in Chapter 3, Section 3.2, also applies to the use of

TABLE 5

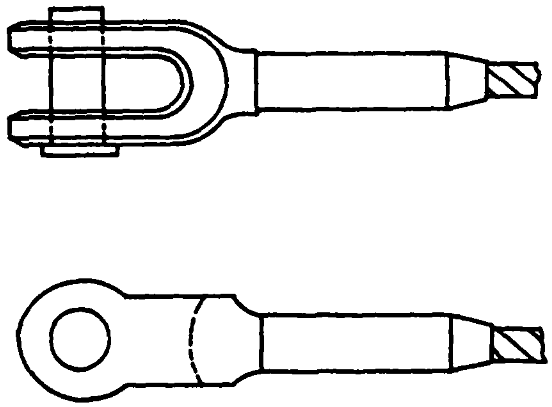
FITTING ASSEMBLY DIMENSIONS

Fitting Size	Cable O.D. 1/32	No. Wires Per Layer	Hole O.D. In Fitting for Conductor Wires	Dual Plug Part No.	Dimensions A	Dimensions B	Dimensions C
1/8	100	12 x 18	3/32	ME-212	1-3/8	Cut to required	7/16
3/16	3/16	18 x 18	1/8	ME-218	1-23/32	length, plus 1" to	1/2
1/4	1/4	15 x 15 18 x 18	3/16	ME-225	1-15/16	1-1/2" additional to allow for	9/16
5/16	5/16	15 x 15 18 x 18 18 x 24 24 x 24	3/16	ME-231	2-5/16	seating of plug.	5/8
3/8	3/8	18 x 18	1/4	ME-237	2-5/8		3/4
7/16	7/16	18 x 18	5/16	ME-243	3-5/16		3/4
1/2	1/2	18 x 18 24 x 24	3/8	ME-250	3-1/2		15/16
9/16	9/16	24 x 24	7/16	ME-256	4-1/4		1-1/8

compressed sleeves. Additional data concerning the use of compressed sleeves will be found in Chapter 13.

3.6 Swaged Terminations

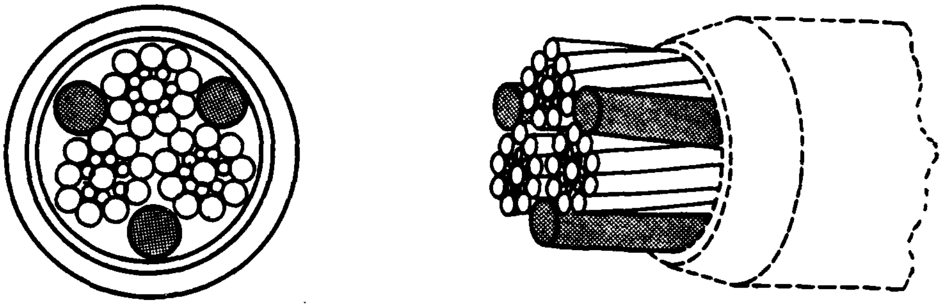
The swaged style of wire termination, possesses a 100% efficiency rating when compared to the breaking strength of the wire being terminated. Its high efficiency is achieved through the use of large hydraulic presses which exert a uniform compressive force on the fitting (Figure 6-6).



SWAGED TERMINATION
FIGURE 6-6

When swaged sockets are used with 3 x 19 wire rope, it is necessary to insert filler pieces into the spaces between the strands, inside the socket, prior to compression of the fitting. The soft wire fillers serve to increase the effective surface area of the 3 x 19 rope and allows a more uniform compression and holding power to be achieved (Figure 6-7).

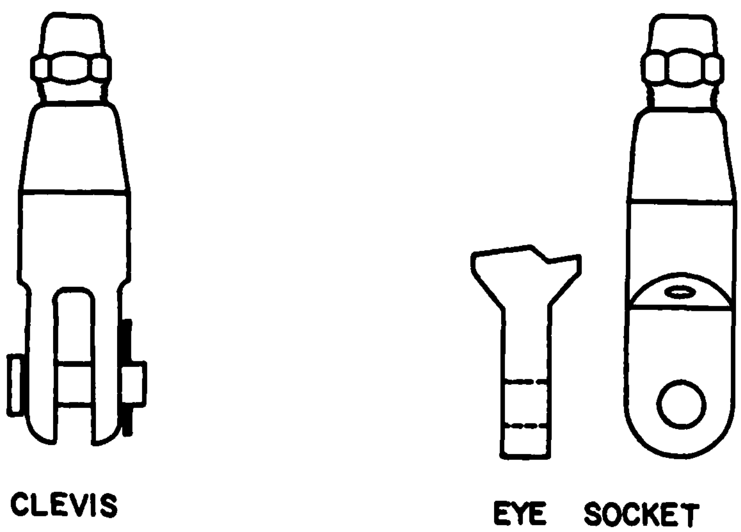
The specialized nature of the swaged termination does not lend itself easily to field applications due to operate the equipment. The frequent retermination of both trawl and hydrographic wires that is required at sea also precludes the use of this particular type of fitting for working cable applications. In addition, the corrosion potential within the swaged socket occurs in an area which is impossible to inspect. Corrosion of the filler wire can occur over time and eventually result in a failure of the termination without prior warning or evidence of weakening.



FILLER WIRES FOR SWAGED TERMINATIONS
FIGURE 6-7

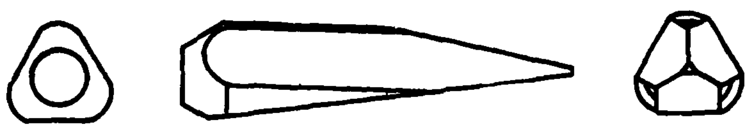
3.7 Mechanical Terminations (Electroline or Fiege)

Perhaps the most common fitting used to terminate the deep sea trawl wire is the Electroline eye socket assembly or "Fiege fitting" as it is frequently called. The Electroline termination (Figure 6-8), is a three component device consisting of a threaded sleeve, socket assembly and a plug which, when properly assembled, will result in a termination strength equal to 95%-100% of the ropes' breaking strength. The high level of fitting efficiency is achieved through the use of the plug as a wedge and by carefully following the assembly instructions.



ELECTROLINE OR FIEGE TERMINATIONS
FIGURE 6-8

Although this type of fitting can be used with all styles of wire rope construction, only three-strand, torque balanced wire rope will be discussed here. Changes in wire construction will require changing the style and possibly the material used to construct the plug (Photograph 1). The particular plug we are interested in for the purpose of this discussion is a triangular plug specifically designed for three-strand wire rope (Figure 6-9).



3X19 WIRE ROPE TRIANGULAR PLUG
FIGURE 6-9

The following assembly procedures have been developed to detail the application of the Electroline mechanical termination for 3 x 19 torque-balanced wire rope. It should be remembered that this application requires that an oversized fitting and special plug be used.

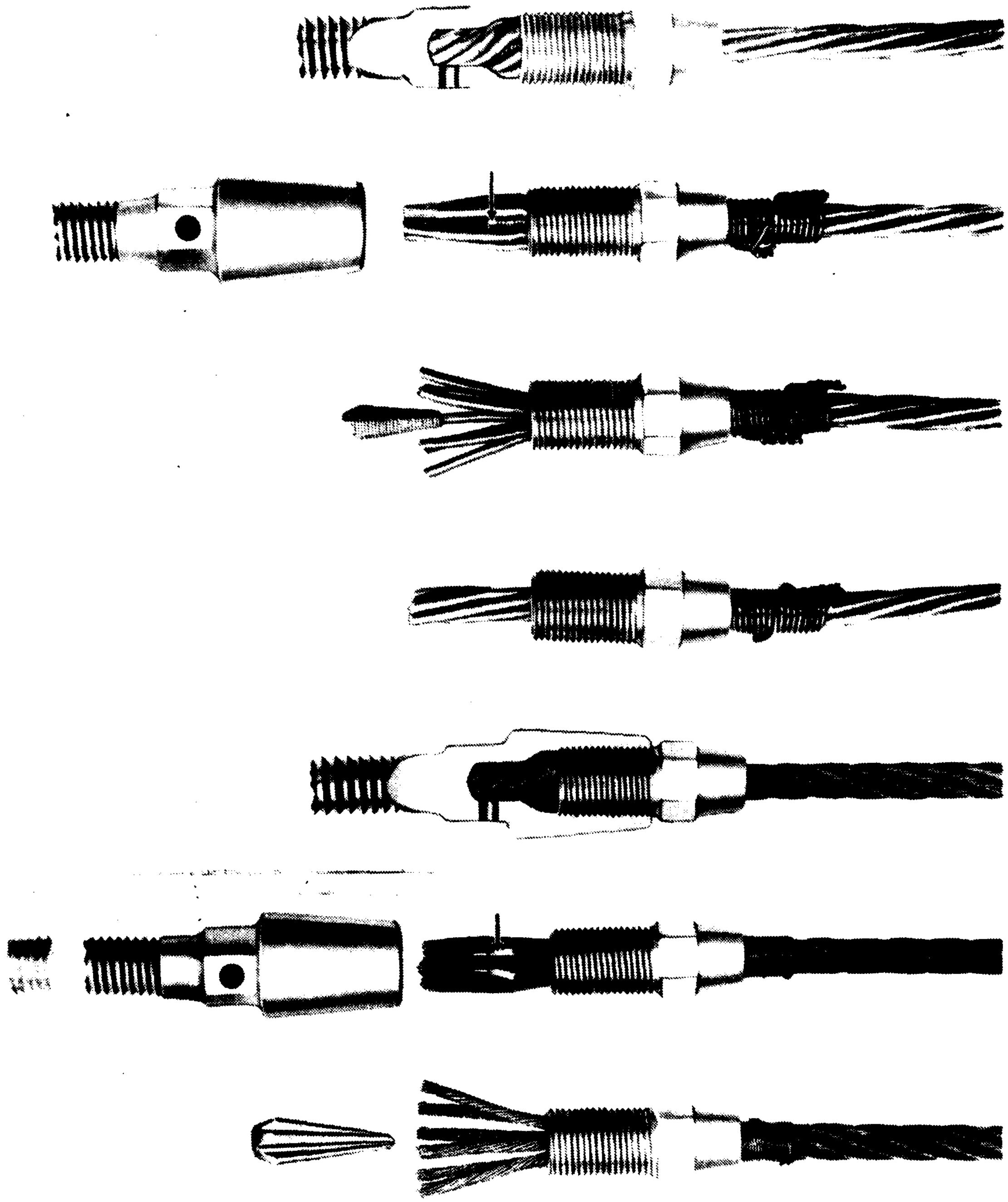
1. When assembling Electroline fittings on wire rope it is recommended that assembly blocks be used to prevent the rope from being nicked by the jaws of the vise, to protect the lay of the rope, and to hold all wires in the strands firmly so the plug can be driven to a solid seat.

   Assembly kits are available in the following three sizes:

Part No.	For Rope Sizes
SP-307A	1/8" thru 9/16"
SP-307B	5/8" thru 1"
SP-307C	1-1/8" thru 1-1/2"

2. The bitter end of the wire to be terminated should be carefully cut to insure a 90° surface. Prior to cutting, the wire should be seized with light wire to prevent unraveling of the rope when cut.

3. Place the Assembly blocks on the rope and place blocks in vise. If the assembly blocks are missing or un-



ELECTROLINE OR FIEGE ASSEMBLIES
PHOTOGRAPH I

available, two short pieces of wood strapping can be used between the vise jaws. The wire length should be carefully measured (Table 4) before securing the wire between the wooden blocks in the vise.

4. Position end of rope to dimension "A" as shown on the chart for the size rope being assembled. Tighten vise firmly. This position of the assembly procedure should be carefully checked with a good ruler to ensure that the exact measurements specified in the chart are met. In addition, care should be taken to ensure that the wire is perpendicular to the assembly block, before tightening vise.

5. Remove seizing on end of rope.

6. Twist threaded end of sleeve over end of rope. Twist in direction of rope lay. Check dimension "B" as shown on Table 4.

7. Unlay one of the three strands. If rope has a right lay, unlay each of the other two strands in counter-clockwise order. If rope has a left lay, unlay each of the other two strands in clockwise order. When done correctly, the three outer strands form a symmetrical basket. Do not attempt to straighten the spiral lay of the three strands.

8. Place the plug in the center of the three strands. Drive the plug downward with a hammer while making certain that each of the strands is positioned properly along the sides of the plug.

9. Once the plug is in position it should be driven to a solid seat using a hammer and draft pin of at least 1/2" diameter. When seated, the top of the plug should be well below the tops of the wires.

10. Remove assembly from vise, remove assemby blocks, and clamp the hex of the sleeve in vise. At this time wrap a layer of tape aruond the wire where it exits the sleeve. The tape should be as close to the base of the sleeve as possible.

11. With a piece of tubing (I.D. of tubing should be 1/32" to 1/16" larger than O.D. of each strand of rope), bend each of the three strands in toward the center of the plug. Tubing is furnished in each of the three assembly kits.

    As an alternative to using the tubing, a small 2" hose clamp can be used to squeeze the ends of the wires toward the center of the plug, allowing the socket to slide over the wire ends and mate with the threaded sleeve.

6-26

12. Place socket over ends of strands, twist on in the direction of the lay of the rope. Engage threads of sleeve and tighten socket securely on sleeve.

During the tightening of the fitting it is recommended that a pair of large crescent wrenchs (20") be used instead of a bar through the eye of the socket. The reasoning behind this is that the bar can and will deform the eye of the socket making insertion of a bushing or screw pin difficult.

13. If assembled correctly, the end of the rope will be visible in the inspection hole. Several threads will be visible on the sleeve below the eye socket after tightening. The best method for checking the visibility of the wires through the inspection hole is with the use of a flashlight. If the ends of the wires are not visible the fitting should be removed, the wire cut, and the termination installed again.

At this time it would pay to check the tape on the wire below the fitting sleeve to determine the amount of slippage that has occurred. The average amount allowable is approximately 1/2".

14. After a proof load is applied to the assembly, the plug will seat further in the sleeve and the rope will not be visible in the inspection hole. This final seating of the plug insures an assembly of maximum strength. After the proof loading of the fitting has been accomplished the fitting should tightened again to recover any exposed threads. The recommended proof load should be 8,000 lbs. for 1/2-9/16', 5/8 wire ropes.

Rope Size	Fitting Rope Size	Sleeve Rope Size	Plug No. -MZ	Dimensions A+1/8 -0	Dimensions B+1/8 -0/8
1/16	1/8	1/8	MZ 1606	1-9/16	5/8
1/8	3/16	3/16	" 1612	1-31/32	3/4
3/16	1/4	1/4	" 1618	2-1/4	13/16
1/4	5/16	5/16	" 1625	2-3/4	1
5/16	3/8	3/8	" 1631	3-1/8	1-1/8
3/8	7/16	7/16	" 1437	3-5/8	1-5/16
7/16	1/2	1/2	" 1443	4-1/8	1-1/2

Rope Size	Fitting Rope Size	Sleeve Rope Size	Plug No. -MZ	Dimensions	
1/2	9/16	9/16	" 2250	4-3/4	1-5/8
9/16	5/8	5/8	" 2250	4-3/4	1-5/8
5/8	3/4	3/4	" 1462	5-1/2	1-7/8
3/4	7/8	7/8	" 2275	6-1/2	2-1/4
7/8	1	1	" 2287	7-7/8	2-3/4
1	1-1/8	1-1/8	" 2299	9-3/8	3-1/8

ELECTROLINE TERMINATION ASSEMBLY DIMENSIONS
TABLE 4

MISCELLANEOUS

The following items are not recorded in any of the literature available to date, but constitute a series of points that can help in detecting or preventing failure, etc.

A. Eye Socket Hole

It has proven to be a good idea to accurately measure the hole diameter of each new fitting prior to its first use and again at the end of each cruise. In this way elongation of the hole due to applied stresses can be detected and the fitting discarded prior to a failure.

Some form of indexing and logging of this data should be established. It is also recommended that when the hole is measured, at least three separate readings be taken to determine an average.

B. Eye Socket Bushing

It is recommended that a busing be used inside the eye socket hole when the pin passing through the eye is smaller than the opening. This approach will lessen the point load exerted on the fitting durig period of high stress.

C. Removal of Fitting and Plug

When a termination is to be removed it is suggested that the wire be cut off as close to the sleeve as possible. The fitting is then disassembled, the sleeve with plug clamped in a vise with the sawn off section of wire uppermost, and the plug and wire driven out with a 1/2" drift pin. It will be noted that a considerable amount of force will be required to accomplish this task.

D. Re-use of Triangular Plug

It is in no way recommended that the triangular plug be re-used after it has been removed from the sleeve. The potential for damage to the plug is very high and re-use can only jeopardize the equipment deployed on the next lowering. Also a careful inspection of the sleeve and eye socket should be made prior to their re-use or storage. Items to look for are broken threads, elongated holes, and cracks in either the sleeve or eye socket.

4.0 ELECTRO-MECHANICAL TERMINATIONS

Since it would be impractical to discuss fully all of the available styles of electro-mechanical terminations this chapter will, instead, concentrate on three specific types of terminations. Specifically these will include: 1) straight mechanical fittings; 2) combination mechanical and epoxy terminations; and 3) helical terminations. It is felt that other termination styles which are available lie somewhere within the range of those discussed in this chapter.

One factor which cannot be stressed often enough is the need to carefully follow the recommended installation procedures for the felting selected. If the termination is to perform as it is advertised it is necessary for the individual installing of the felting to understand that recommended procedures should not be compromised by a lack of attention to detail. Required wire lengths should not be estimated and the necessary curing times for epoxies must not be disregarded in an attempt to redeploy an instrument. The result of such careless practices is usually the loss of an expensive piece of equipment.

4.1 Electroline E-M Cable Terminations

This style of termination, like its wire rope counterpart, relies on the use of a series of plugs to achieve its holding power. Because this is a purely mechanical termination it can be used immediately after assembly and testing. Other terminations which combine an epoxy filler in the mechanical fittings are restricted by the curing time of the epoxy and therefore have a relatively long time period between assembly and use.

The purely mechanical termination is quite simple to install in the field requiring only basic hand tools and no special technical training on the part of the installer. The only critical aspect of assembling this style of fitting is close attention to the installation instructions. By carefully following the instructions detailed below the maximum efficiency of the felting can be assurred.

4.2 Installation Procedures

The following procedures have been developed for the termination of double armor electro-mechanical cable. The success of this termination rests on the proper installation of two hollow plugs (Figure 6-10) and the careful measurement of the wire lengths required in Table 5.

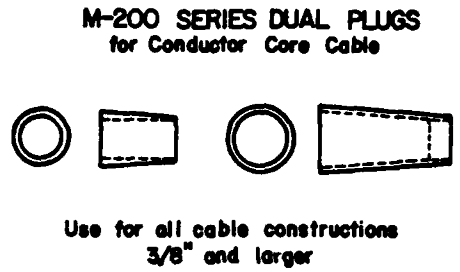


FIGURE 6—10

Installation of the fitting can be accomplished using the following procedure:

1. The cable to be terminated should have the better end seized and the cable cut off square. Once this is accomplished place the cable in the assemby block, position the end of the strand to dimension A and B for the size cable being terminated and clamp the assembly block in a vise (Figure 6-1). When the cable is clamped in the vise the seizing can be removed.

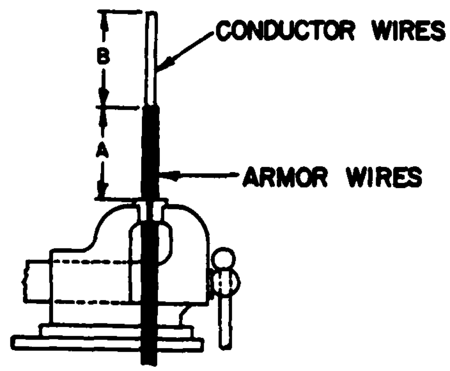


FIGURE 6—11

TABLE 5

FITTING ASSEMBLY DIMENSIONS

Fitting Size	Cable O.D. . 1/32	No. Wires Per Layer	Hole O.D. In Fitting for Conductor Wires	Dual Plug Part No.	Dimensions		
					A	B	C
1/8	100	12 x 18	3/32	ME-212	1-3/8	Cut to required	7/16
3/16	3/16	18 x 18	1/8	ME-218	1-23/32	length, plus 1" to	1/2
1/4	1/4	15 x 15 18 x 18	3/16	ME-225	1-15/16	1-1/2" additional space to allow for	9/16
5/16	5/16	15 x 15 18 x 18 18 x 24 24 x 24	3/16	ME-231	2-5/16	seating of plug.	5/8
3/8	3/8	18 x 18	1/4	ME-237	2-5/8		3/4
7/16	7/16	18 x 18	5/16	ME-243	3-5/16		3/4
1/2	1/2	18 x 18 24 x 24	3/8	ME-250	3-1/2		15/16
9/16	9/16	24 x 24	7/16	ME-256	4-1/4		1-1/8

2. Twist the threaded sleeve over the end of the cable and check the length of the exposed armor strand against dimension C in Table 5. If this length is compatible with the table proceed to unlay the outer armor wires. However, if this dimension is not met reposition the cable in the blocks before continuing. Place the large hollow plug over the center wires of the cable and carefully drive it to a solid seat (Figure 6-12). Once seated the inner armor wires should be unlayed and the small hollow plug slipped over the conductor wires and driven to a solid seat.

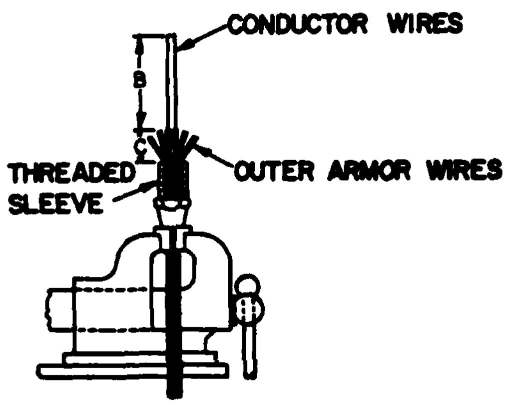


FIGURE 6-12

3. Remove the assembly blocks from the vise and the cable and clamp the threaded sleeve in the vise as shown in Figure 6-13. The broomed-out armor wires can now be bent inward around the conductor wires. In order to protect the conductor wires during the process, a piece of tubing should be slipped over the conductors prior to bending the armor wires inward.

Once this step is completed twist the socket or clevis end portion of the fitting over the ends of

the wires and feed the conductor wires through the hole provided. Engage the threads of the sleeve and tighten securely. When tightening the socket or clevis it is advisable to remember the procedure discussed in section 3.7(12).

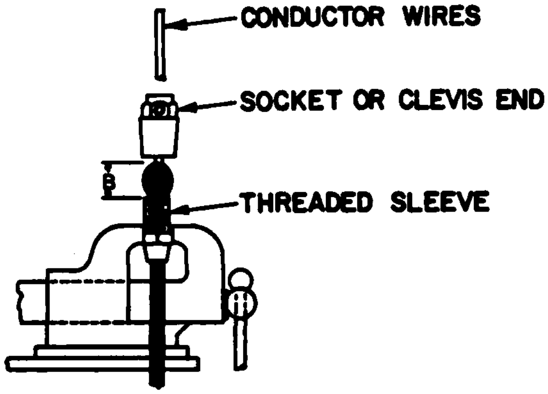


FIGURE 6-13

4. Once the socket or clevis has been securely tightened on the sleeve the ends of the armor wires or tubing will be visible in the inspection hole. If the intsallation has been properly accomplished (Figure 6-14), several threads will be exposed on the sleeve. Should the armor wires or tubing not be visible in the inspection hole the termination should not be used, but should instead be removed and the cable reterminated.

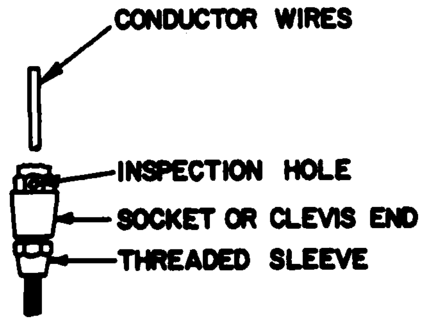


FIGURE 6-14

5. It is advisable after the termination has been successfully installed to perform a pull test at a load equal to at least two times the anticipated instrument load to assure the safety of the equipment.

4.3 Combination Mechanical/Epoxy Termination

The particular fitting which will be discussed here is the DYNA-GRIP termination for electro-mechanical cables. The basic termination design consists of an oval-shaped hollow insert which slips over the cable, a set of helically-formed rods which wrap (by hand) over the cable and insert, a housing with internal contour to match the rods and insert and a threaded retainer (Figure 6-15). Holding strength of the termination is develped by the preformed principle of the helically formed rods and gripping by the matching insert and housing (Figure 6-16). There is no reliance upon special tools or user proficiency. Every installation is uniform and repeatable in holding ability and appearance. There is no crushing or deformation of the cable elements, and yet, the termination will hold the full-rated strength of the cable for which it is designed.

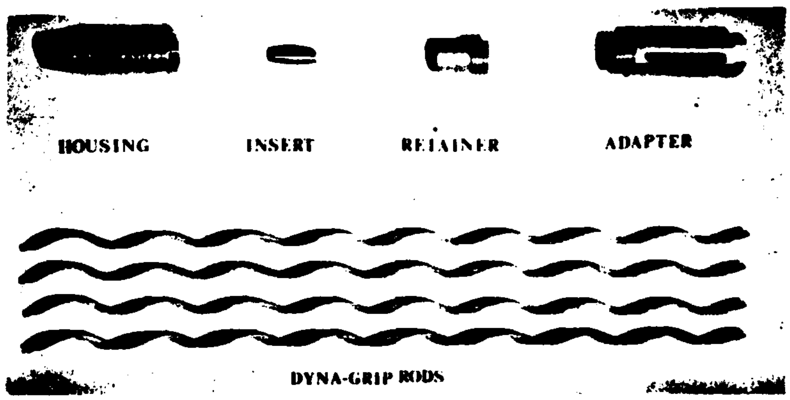


FIGURE 6–15

1. Cable Preparation: If the armor is to be terminated at the fitting, care should be taken to allow for a sufficient length of armor to extend beyond that of the Rods. The outer armor wires should be cut about 1/4" shorter than the inner wires and then taped. After application of the Dyna-Rods, the inner armor should be bent away from the cable to prevent chaffing of the insulation.

2. For proper positioning of the helical rods (Figure 6-17), match the center mark on the insert to the color mark on the rods. For proper and easy installation of the rods, the following are important:

   a. Begin application with a two-rod subset and end application with a two-rod subset.

   b. Wrap the rods one subset at a time about the cable and over the insert starting at the trimmed end of the cable.

   c. Align the ends of the rods closely with each other.

   d. Do not allow gaps between subsets or accumulation of gaps between subsets or accumulation of gaps could interfere with application of subsequent subsets.

   e. Do not allow any of the helical rods to cross each other.

3. After complete set of rods is wrapped on (Figure 6-18), slide housing over rod and insert assembly until it seats over insert.

4. Insert retainer and screw tightly into housing. Use the clevis with hex keys as a spanner wrench (Figure 6-19). When retainer is properly in place, there will be no looseness in the entire assembly.

5. Place the housing in a nose down position. The epoxy filler is then prepared.

   Preparation of epoxy:

   1. Thoroughly mix the contents of each can before combining them. Each part tends to separate into layers. Each component must be thoroughly mixed until it is homogeneous.

      This is critical to proper cure of the material.

      Combine the contents of the two cans. Mix the combination thoroughly for five to ten minutes. The epoxy is then ready to pour.

6. Dam the leading edge (nose) of the housing. A mastic modeling clay or similar material can be used. Pour the epoxy material into the housing. It is important that the epoxy penetrates into the interstices between

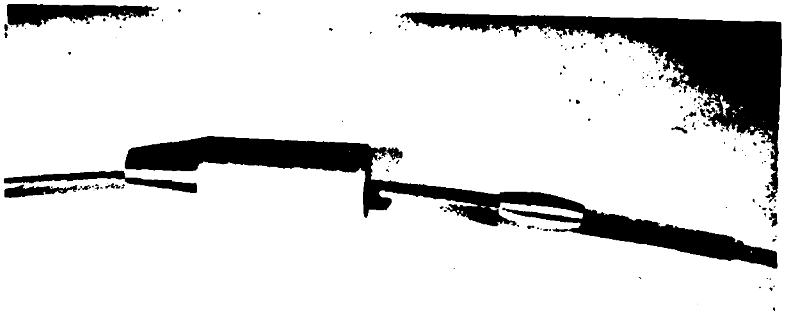


FIGURE 6-16

FIGURE 6-17

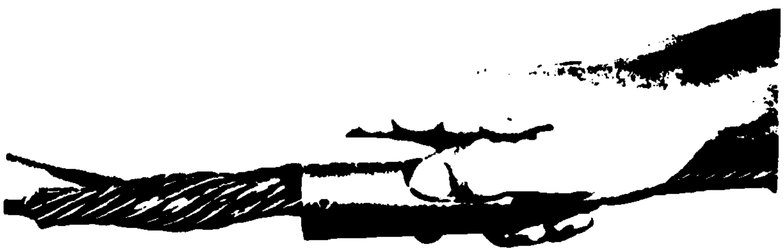


FIGURE 6-18

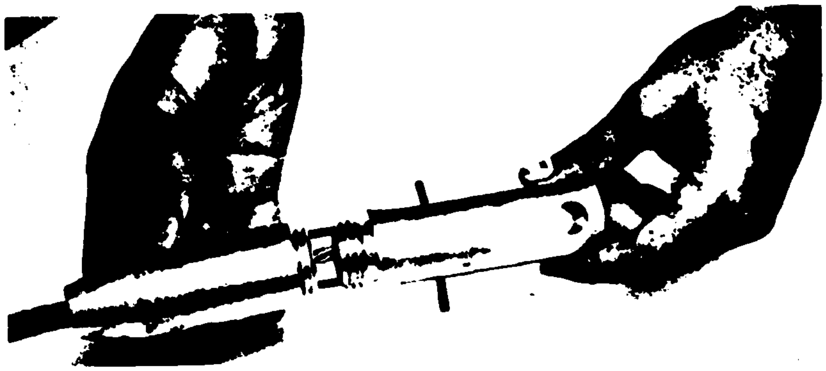


FIGURE 6-19

FIGURE 6-20

FIGURE 6-21

individual armoring rods. Penetration of the adhesive may be checked by breaking the dam and visually checking to see the adhesive seeping through the fitting.

7. Screw in mounting adaptor and lock thread with groove-pin (or set screws) before the epoxy filler hardens (Figure 6-20).

8. Completed assembly (Figure 6-21).

9. The following techniques should be observed in order to termination:

   a. Do not terminate the armor wires of electro-mechanical cable inside the housing or apply the epoxy filler in a manner which would cause a bond of the cable armor to the electrical core.

   b. Allow the epoxy filler material to cure for 24 hours (normal temperatures) before using.

   c. Reuse of the preformed helical rods is not generally recommended after load has been applied.

   d. Caution should be used when overboard sheave has a groove diameter less than 40 times the diameter over the helical portion of the assembly.

4.4 Helically Wound Terminations

This particular style of termination is designed for electro-mechanical cables larger than 1 inch diameter (Figure 6-22). Certain limitations are present in the use of this termination and are expressed below.

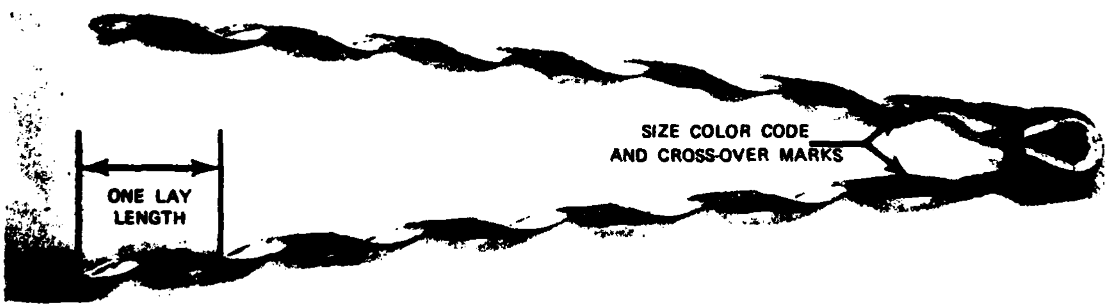


FIGURE 6-22

1. The preformed Cable Stopper must have the same lay direction as the cable. CAUTION: Never use a preformed Cable Stopper with an opposite lay cable.

2. In the application of preformed Cable Stoppers, marlin spikes or screw drivers should be used only as an aid in splitting the legs and snapping the ends in place.

3. Wire rope thimbles or equivalent fittings of the same size as the Cable Stopper should be used.

Successful application of this termination can be achieved by following the procedure detailed below.

1. Start application by wrapping on two lay lengths of first leg, starting at cross-over marks (Figure 6-23).

FIGURE 6-23

2. At this point of the application, install a heavy duty wire rope thimble (if required) (Figure 6-24). If the eye is intended to be attached by a shackle, it should then be placed on a pin or shackled to a pad eye to keep it from turning. Match the cross-over marks and apply the second leg the same lay length as the first. Application is made easier if the leg of section being applied is pulled out and around the cable in one continuous motion.

FIGURE 6-24

3. Use a marlin spike or screw driver and split the legs into sections for the one pitch length as shown in Figure 6-25.

FIGURE 6–25

4. Split both legs back to the applied portion of the Cable Stopper as shown in Figure 6-26. The section closest to the cable (#1, above) should be applied and followed consecutively by the sections closest

FIGURE 6–26

5. At this point the first section should be applied to completion, followed consecutively by each of the other legs (Figure 6-27).

FIGURE 6–27

6. As shown in Figure 6-28, a marlin spike or screw driver can be used to snap the ends into place.

FIGURE 6—28

7. Completed application of the preforemd Cable Stopper (Figure 6-29). Make sure all rod ends are snapped into place and that all rods are in contact with the cable. Should there be one rod under another, remove to that point and re-apply.

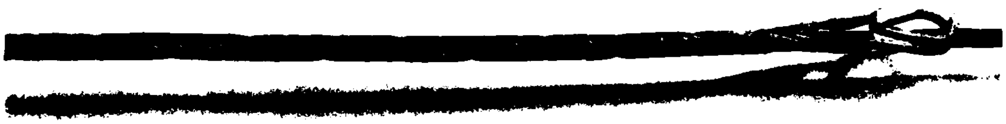


FIGURE 6—29

8. The following should also be considered when this type of termination is selected:

   a. The preformed Cable Stopper dead-end was developed as a Strain-relief fitting for data cables. These can be applied to either bare armor or jacketed cables and are normally designed to hold 50% of the rated breaking strength.

   b. The eye of the grip is made with sufficient length to allow the cable to proceed on through and beyond the termination point. This provides complete continuity of the cable to an overboarding application with a strain-relief member of proven performance.

c. The preformed helical concept is the only method of termination or holding of a cable that will not crush the cable or create a high-stress potential failure point.

CHAPTER 7

Equipment Lowering Mechanics

H.O. BERTEAUX

1.0 INTRODUCTION 7-2
2.0 MECHANISMS CAUSING LOWERING CABLE DAMAGE 7-2
2.1 STATIC AND QUASI-STATIC TENSILE LOADS 7-3
2.2 WAVE INDUCED DYNAMIC LOADS 7-4
2.3 ZERO LOAD 7-6
2.4 SNAP LOADS 7-7
3.0 PREDICTING CABLE LOADS 7-8
3.1 IMMERSED WEIGHT STATIC LOAD 7-8
3.2 HYDRODYNAMIC RESISTANCE QUASI-STATIC LOAD 7-8
3.3 TERMINAL VELOCITY-ZERO LOAD 7-12
3.4 VIRTUAL MASS-INERTIA LOADS 7-13
3.5 ALL FORCES CONSIDERED-STEADY STATE PEAK TENSIONS 7-15
3.6 SNAP LOADS 7-19
3.7 ADVANCED CABLE DYNAMICS 7-23
4.0 RECOMMENDATIONS 7-25
4.1 EQUIPMENT DESIGN CONSIDERATIONS 7-25
4.2 EQUIPMENT HANDLING CONSIDERATIONS 7-27
REFERENCES 7-29
BIBLIOGRAPHY 7-30

1.0 INTRODUCTION

Improper design considerations while lowering or retrieving equipment at sea can result in substantial cable damage including rupture followed by the total loss of a valuable payload.

The hydrodynamic behavior of the payload at the end of the lowering cable is often unknown or ignored. Yet the shape and weight of the payload has considerable bearing on the cable performance. For example, a payload with a large drag area can substantially add to cable static tension when the cable is hauled in. Conversely, while paying out the payload may fall slower than the cable itself thereby creating a slack cable condition. This slack condition may result in a subsequent snap load, when the lowering is stopped and the payload impacts the cable, or in a kink in the wire, or both. Payload spinning, kiting and tumbling can also obviously impair an orderly lowering.

Recurrent causes of cable damage include: loading the cable beyond its yield point (or breaking strength), fatigue failure, and kinking. In this section the mechanisms which create these detrimental conditions are first reviewed. These potential failure mechanisms include: quasi-static tensile loads, wave induced dynamic loads, zero load (slack cable), and impact (snap) loads.

The mathematical concepts which are used to predict and quantify these causes of cable failure are also reviewed and their use illustrated with a few typical examples.

In the last part of this chapter specific design recommendations are made for improving the payload hydrodynamic behavior. Finally, the operational limits such as maximum length of cable paid out or allowable payout rates are discussed.

2.0 MECHANISMS CAUSING LOWERING CABLE DAMAGE

One way to obtain measurements of oceanographic parameters at great depths is to lower sensing instrument packages with electromechanical cable. Of necessity these cables are kept to small, workable sizes but because of the long lengths deployed their immersed weight often results in very high tension levels. Vessel motion, due to wave action, introduces additional cyclic loads which can, and often do cause, cable deterioration due to flexure fatigue. Kinks can occur in the cable during zero load conditions. These problems often result, at best, in loss of electrical signal due to short or open circuits and at worst in complete failure of the cable and total loss of the instrument package.

While hanging free from the ship, the tension in the cable is the sum of the static load, due to cable and instrument immersed weight, and the dynamic load, due to cable and attached equipment inertia and hydrodynamic resistance. Most of the time static and dynamic effects occur simultaneously. However, for the sake of clarity it will be helpful to consider them separately.

2.1 Static and Quasi-Static Tensile Loads

By and large, cables are designed and built to resist fair amounts of tensile loads. As their working life progresses their original strength is reduced by corrosion, abrasion and normal wear and tear. If the tensile loads come close to the actual strength of the cable, permanent cable damage or even total failure will occur. To prevent this form of failure it is necessary to understand and quantify the mechanisms of tensile loading.

The first factor of tensile loading is plain weight. The weight that the cable must support at its ship end is made of two parts: 1) the weight in water of the payload and 2) the weight in water of the cable itself. Whereas the payload immersed weight remains constant, the immersed weight of the cable increases with the length of cable paid out. In many situations the payload weight is but a small fraction of the cable weight.

The second factor of cable tensile loading is due to hydrodynamic resistance (drag). If, on a calm day, the lowering winch is turning at a constant rate, the resulting steady state motion of the payload and lowering cable through the water will produce a quasi-static loading which, depending on the direction of motion (up or down), will add to or subtract from the static loading due to cable and payload weight. A few words on the nature of hydrodynamic resistance will help understand how cable and payload drag interact and combine to drastically change the static loading due to weight only.

Simply stated, hydrodynamic resistance is the force experienced by a body when moving through a fluid. This resistance is due to a combination of viscous and pressure effects. These two effects are concurrent. Their relative magnitude depends, however, on the nature of the flow past the body. As long as the flow remains smooth, or laminar, shear stresses predominate and the resistance, or drag, is esentially due to the friction of the fluid on the bodies immersed surface (skin friction drag). On the other hand, when a combination of fluid speed and body shape (blunt bodies) result in a wake past the body the drag force is then essentially due to the pressure difference between the upstream and downstream sides of the body (pressure drag).

To illustrate the point, the force needed to tow a small but long and neutrally buoyant fishing line aft of a sail boat is essentially due to friction drag on the line. On the other hand the force experienced by someone towing a fully submerged bucket from a short rope aft of the same boat is essentially pressure drag.

In applications involving lowering and hauling equipment to and from the sea floor it is fair to say that the hydrodynamic resistance experienced is the sum of the friction drag on the cable and of the pressure drag on the equipment or payload at the end of the cable.

Cable drag is directly proportional to cable length, whereas equipment drag remains essentially constant. When hauling in, the hydrodynamic resistance will increase the static load due to cable and equipment weight. Its maximum contribution of course is at the beginning of the haul when the cable is longest. Methods to calculate drag forces are reviewed in the next section.

2.2 Wave Induced Dynamic Loads

Next we will consider the dynamic loads imparted to the cable as the ship heaves, rolls, and pitches in rough seas.

After lowering the payload to a certain depth (say 2000 meters) let us secure (stop) the winch. If the cable tension at the head sheave could then be read and displayed, the record would show large fluctuations around a mean. This mean would of course be the immersed weight of the cable paid out and attached equipment. Deviations from the mean are due to dynamic forces imparted on the cable by the motion of the head sheave. As the cable and attached equipment are pulled towards the surface or allowed to plunge back into the sea the cable and the equipment experience both drag and inertia forces.

As previously mentioned, the drag forces are caused by cable and payload instantaneous speed. The inertia forces are caused by cable and payload instantaneous change of speed. Both forces are concurrent. Drag forces reach a maximum when the speed is largest, inertia forces are greatest at the time of maximum acceleration - usually when cable and payload are at rest, at the beginning of a new motion cycle. Here again it may be instructive to briefly look at the nature of the inertia forces. If at some instant the cable and the equipment are hanging still from the ship (zero speed) and at some later but proximate instant cable and payload are pulled upwards at some speed by a ship roll, the tension at the sheave increases. This tension increase is caused by the "inertia" of the cable and equipment which "resent" and resist the instantaneous upward pull.

In general the inertia force can be defined as the force required to change the speed of a body. Its magnitude equals the product of the body mass by the change of speed experienced per unit of time (acceleration).

Fully immersed bodies do trap and entrain a certain amount of water in their motion. This entrained water undergoes the same acceleration as the body itself. The effect is as if the mass of the body had been increased. In fact the actual mass to be accelerated, called the body virtual mass, is the sum of the body mass and of the mass of the entrained water. As a result of the increase in mass, the force needed to accelerate a body in water may be much larger than in air. For example the starting load to accelerate an elevator from rest would be much larger if the elevator was fully submerged (neglecting buoyancy effects). Formulas to calculate inertia forces are presented in the next section.

Now let us go back to our ship and let the winch run again, hoisting the equipment back to the surface at some constant hauling speed. The hydrodynamic resistance due to this additional speed will, at least in the beginning when cable weight reduction is not significant, increase the tension mean and therefore also the tension peaks previously experienced when heaving to, with the winch secured.

The instantaneous tension is now the algebraic sum of four simultaneously occurring effects, namely:

- the immersed weight
- the drag due to hauling speed
- the drag due to wave induced motion
- the inertia forces to accelerate (or decelerate) the cable and the equipment.

This time varying, wave induced, tension results in cyclic stresses which can cause the wires and/or the conductors of a cable to fail in fatigue.

It is a well known fact that the number of fatigue cycles to total failure dramatically decreases as the cyclic tension increases. In instrument lowering applications, because of the long lengths of cable required, the tension can reach a very large fraction of the cable strength. Under these conditions, only a few hundred cycles of repeated stresses can severely damage the cable (See Reference 1).

Keeping the wave induced loads and their time of application small will prevent accelerated fatigue cable deterioration.

2.3 Zero Load. Slack Conditions

Zero load can be the prelude to catastrophy. A slack cable can easily jump out of a sheave, can kink, or it can be subjected to severe snap loading. The payloads attached at the free end of the cable may force the cable to unlay and turn on itself. If the cable is allowed to become slack at some later time it will relieve some of the stored torsional energy by forming one or a number of twisted loops at the point of slack. When tension is reapplied the loops are pulled tight, the armor wires and the conductors are then severely bent thus permanently damaging the cable at the point of kink.

Understanding the mechanisms leading to slack conditions is a first step towards the prevention of their occurrence.

If a body heavier than water is allowed to free fall to the sea floor, it will first accelerate and gain speed. As speed increases so does the hydrodynamic resistance on the body. Sooner or later the drag will equal the pull of gravity and the body will continue to fall at a constant maximum speed called "terminal velocity." This being accepted, let us consider what happens as the equipment is lowered to the bottom.

Assuming the sea to be flat calm and the winch to pay out at some constant and reasonable speed, then the equipment will descend smoothly at the payout speed. But if this speed is increased beyond the equipment's own terminal velocity then the cable will override the equipment and form a slack loop probably full of kinks.

One might be tempted to think that paying the cable at a rate less than the equipment terminal velocity would prevent slack conditions to occur anywhere along the cable. This is not always true. As evidenced in an example presented in the next section, in certain cases a length reached where the combined drag on the cable and the equipment entirely negates the gravity pull. The cable will then again become slack, this time at the shipboard end.

Now let us assume a situation where the winch is secured but the ship is rolling heavily. On a down roll the head sheave may well reach speeds high enough to momentarily create slack conditions either at the sheave, or at the equipment end, or at any point in between. Of course such high speeds can also be obtained when paying out from a rolling ship.

Methods for determining conditions of zero tension and points of occurrence will be briefly reviewed in the next section.

2.4 Snap Loads

Cable tension, as we have seen, is the algebraic sum of the external forces acting on the cable, namely the static force due to weight and the dynamic forces of inertia and hydrodynamic resistance.

This dynamic force can be either compressive or tensile. When the compressive component exceeds the static tensile force the cable goes slack. The payload is then allowed to travel on its own until the cable catches it again. Severe snap loads, as high as ten times the immersed weight of the payload (Reference 2) are then imparted to the cable.

It may again be instructive to describe the mechanism which produces snap loads in some detail. Let us assume that an up roll is pulling hard and fast on the cable. The steel cable is rather stiff, having a high modulus of elasticity. The payload has a large virtual mass. It is heavy and its ugly shape entrains a lot of water. The upper end of the cable moves with the ship. Because of its inertia the payload does not move appreciably yet. The cable is forced to stretch and because of its stiffness the pull on the payload increases at a rapid rate. As a result the payload starts to move faster acquiring upward speed and momentum.

Now comes the down roll. The pull of the cable on the payload diminishes and vanishes as soon as the distance between payload and cable shipboard end equals the relaxed (no load) length of the cable. The payload then starts to travel on its own. It still has considerable momentum and keeps on going upwards, slowing down until the pull of gravity stops it. It then reverses direction of motion. It starts to fall acquiring downwards speed and momentum.

In the meantime the cable is still going down following the ship down roll and giving plenty of time for the payload to gain considerable downwards momentum. Now comes the next up roll. The cable rushes back to the surface. When the distance between the upper end of the cable and the payload position again equals the unstretched length of the cable, the cable starts to pull on the payload. The great force necessary to rapidly stop and reverse the direction of the payload constitutes the snap load.

If properly timed, that is if the wave frequency is such as to permit the procedure to repeat itself the cable will be subjected to a series of snap loads and probably will break.

A simple mathematical model to predict the occurrence of snap loads and quantify their magnitude is presented in the next section.

3.0 PREDICTING CABLE LOADS

This section will present the formulas and certain simple analytical methods which will permit a reasonable prediction of both the static and dynamic cable loads.

3.1 Immersed Weight. Static Load

The weight of a fully immersed object equals the weight of the object in air less the weight of the water displaced by the object. If the two weights are equal the object is said to be neutrally buoyant. If the air weight of the object is less than the weight of the water displaced the buoyant object will want to come back to the surface.

Example 7.1

What is the static load at the ship due to 2000 meters of 1/2 inch 3x19 wire rope supporting a cylinder of cast iron 4 feet high by 2 feet in diameter.

Use:	Weight in water of 1/2" 3x19	= .341 lb/foot*
	Water density	= 64 lbs/cu foot
	Cast iron density	= 480 lbs/cu foot
	One meter	= 3.28 feet

Solution

Air weight of cylinder	=	π x 4x450	= 5655
Weight of water displaced	=	π x 4x64	= - 804
Immersed weight of cylinder			= 4851
Immersed weight of cable	=	2000x3.28x.341	= 2237
Static load at ship end of cable			= 7088 lbs.

(*Data by U.S. Steel Corporation)

3.2 Hydrodynamic Resistance - Quasi-Static Load

The hydrodynamic resistance of a fully submerged object moving at a constant speed "V" (ft/sec) can be estimated using the formula:

$$D = 1/2 \rho\, C_D\, AV^2 \qquad (7.1)$$

where D is the hydrodynamic resistance or drag (lbs)

ρ is the water mass density = 2 slugs/cu.ft.

C_D is the drag coefficient

A is the object area used to empirically derive the drag coefficient (sq-ft)

Pressure drag coefficients for various body shapes (spheres, cylinders, plates, etc...) have been widely published in the literature (References 3 & 4). Longitudinal drag coefficients for cables and long cylinders have also been extensively studied. Published values vary from .02 for rough cylinders to .0025 for smooth cylinders (See Figure 7.1).

The following example illustrates the use of formula (7.1).

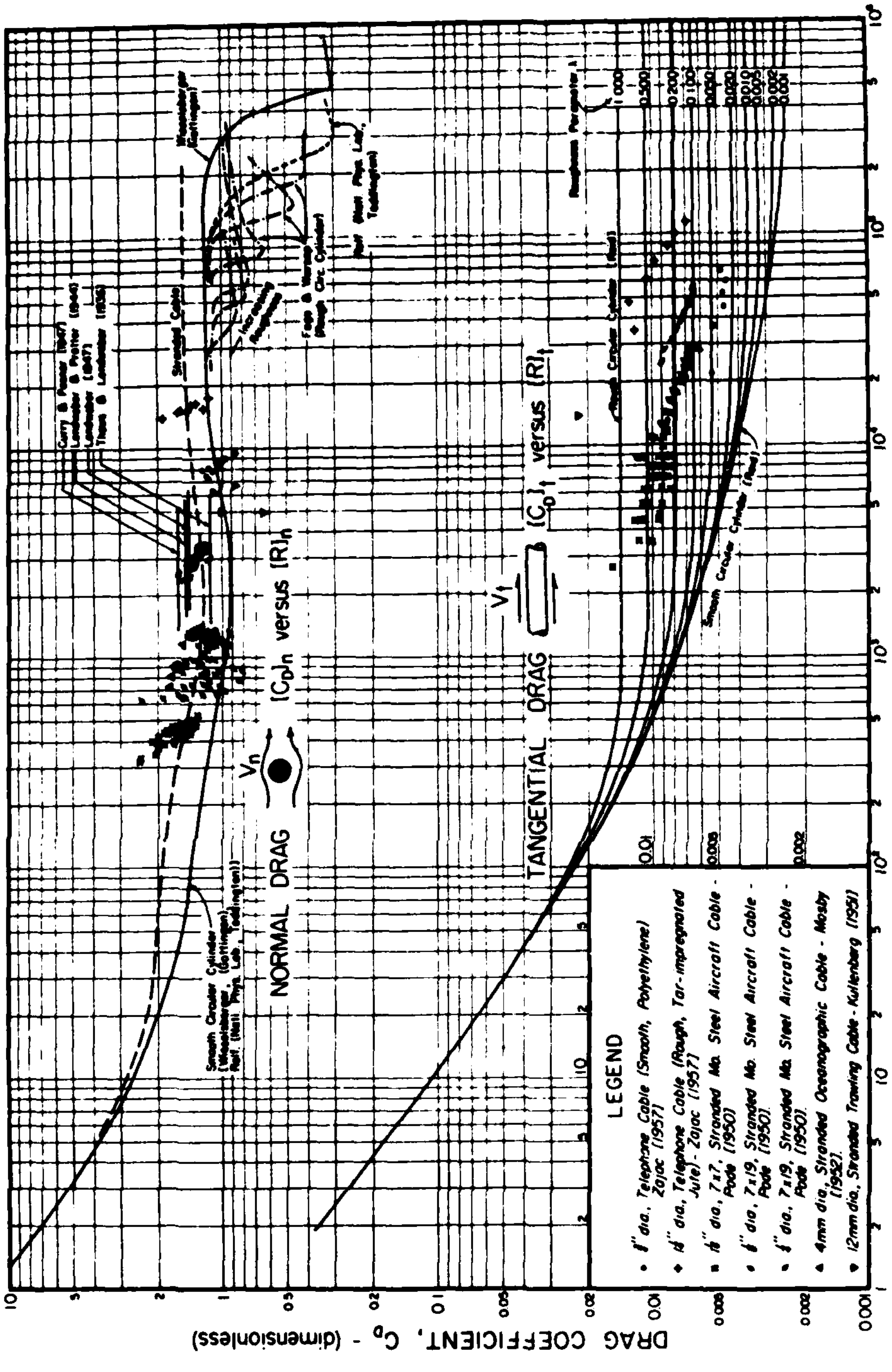
Example 7.2

A biological sampler is lowered to a depth of 2000 meters with the help of a 3/8 inch 3x19 wire rope. It is then hauled back at a constant speed of 100 meters/min. Find the tension at the upper end of the cable immediately after the starting transient, using the following characteristics:

Immersed weight of cable	=	.191 lb/ft*
Drag coefficient of cable	=	.01
Shape of sampler	=	cone, base up, filled with water
Diameter = 6 ft		Height = 5 ft
Immersed weight of sampler = 200 lbs		Dry weight = 320 lbs
Drag coefficient of sampler	=	1.0

Also find the percent increase due to drag over the plain static load.

(*Data by U.S. Steel Corporation)

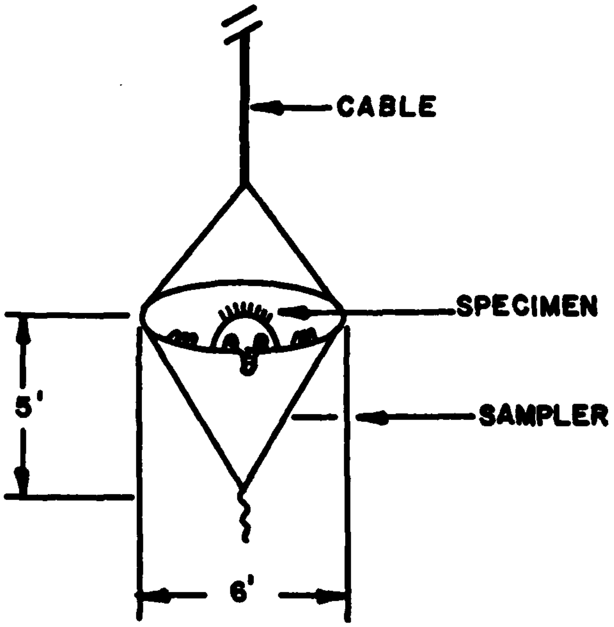


From "Buoy Engineering" by H. O. Berteaux. (Wiley And Sons) Page 101

Drag Coefficient versus Reynolds Number for flows normal and tangential to smooth and rough circular cylinders.

FIGURE 7-1

Solution



Total immersed weight of cable	=	.191x2000x3.28	= 1253 lbs
Weight of sampler	=		200 lbs
Static tension	=		1453 lbs
Hauling speed	=	100x3.28/60 = 5.46 ft/sec	
Skin area of cable	=	$\pi \times \frac{3}{8} \times \frac{2000}{12} \times 3.28$	
	=	644 sq-ft	
Cable drag	=	0.01x644x5.46x5.46 = 192 lbs	
Cross section of sampler	=	πx3x3	
	=	28.3 sq.ft.	
Sampler drag	=	1.0x28.3x5.46x5.46 = 844 lbs	
Total drag	=	1036 lbs	1036 lbs
Tension at cable upper end	=		2489 lbs
Percent increase due to drag	=	1036/1453 = .713 or 71.3%	

If "s" is the length of cable paid out, the quasi-static tension T(s) at the ship end can be found using the following expression:

$$T(s) = W_p + W_L s + \tfrac{1}{2} \rho C_c \pi D_c s V|V| + \tfrac{1}{2} \rho D_p A_p V|V| \qquad (7.2)$$

where

W_p = immersed weight of payload (lbs)

W_L = immersed weight of cable per unit of length (lbs/ft)

C_c = cable longitudinal drag coefficient

D_c = cable diameter (ft)

C_p = payload normal drag coefficient

A_p = payload normal cross section (sq-ft)

V = constant cable speed, hauling being positive and lowering negative

$|V|$ = absolute value of V

Cable drag is both a function of speed and length. If the amount of cable paid out and the speed of lowering are large enough, the combined cable and payload drag can become as large as the cable and payload immersed weight. The tension in the cable then becomes zero.

For a given lowering speed V, the length of cable necessary to produce a slack condition can be found by setting T(s)=0 in (7.2) and solving for s.

3.3 Terminal Velocity. Zero Load

At terminal velocity the immersed weight of the object "W" equals the drag on the object, a condition which is expressed by:

$$W = \tfrac{1}{2} \rho C_D A V_T^2$$

Therefore the terminal velocity V_T of the object is given by:

$$V_T = \sqrt{\frac{2W}{\rho C_D A}} \qquad (7.3)$$

Example 7.3

Find the terminal velocity of:

1. The biological sampler described in Example 7.2 using a nose down drag coefficient $C_D = 0.2$
2. The 2000 meters of 3/8" 3x19 wire rope combined with the sampler.

   Use $\rho = 2$ slugs/cu.ft

Solution

1. Terminal velocity of the sampler.

   From previous computations,

Immersed weight of sampler = 200 lbs

Cross section = 28.3 sq-ft

Terminal velocity $= \sqrt{\dfrac{200}{0.2 \times 28.3}} = 5.94$ ft/sec

or 109 meters/min

2. Terminal velocity of cable and sampler combined.

   From previous computations,

Immersed weight of cable = 1253 lbs

Skin area of cable = 644 sq ft

Drag coefficient of cable = 0.01

Terminal velocity $= \sqrt{\dfrac{1253+200}{644 \times 0.01 + 0.2 \times 28.3}} = 10.95$ ft/sec

This example shows that a payout rate in excess of 109 meters/min would cause a slack condition in the wire rope lower end. Similarly a downwards speed in excess of 11 ft/sec - which could be easily obtained by a combination of payout rate and ship down roll - would produce a slack condition at both ends of the 2000 meters length of cable.

3.4 Virtual Mass. Inertia Load

In order to accelerate a body immersed in water not only must the body be accelerated but also a certain amount of water close to or ahead of the body. As a result the force F' needed to accelerate the body in water is greater than the force

F required to accelerate the same body in vacuum. This can be expressed by:

$$F' = (m+m')a > F = ma$$

where m is the body mass

m' is the added mass of the entrained water

a is the acceleration.

The added mass is usually computed using

$$m' = c_m \rho \,(Vol)$$

(7.3)

where C_m is the added mass coefficient, ρ is the water mass density (slugs/ft^3) and Vol is the volume of water displaced by the immersed body (cu ft)

Added mass coefficients for bodies of different shape (sphere, cylinders, plates, etc...) have been empirically determined for linear and oscillating accelerations. Published values of added mass coefficients pertinent to cable lowering problems can be found in Reference 4.

The virtual mass m_v is the sum of the body mass m and of the added mass m'.

$$m_v = m + m'$$

Example 7.4

1. Find the virtual mass of the biological sampler previously discussed.

2. Find the inertia force on the lower end of the cable and the percent increase over the static load at that end if the sampler is accelerated towards the surface from rest to a speed of 8 ft/sec (146 meters/minute) in a) 8 seconds, b) 2 seconds.

Use = 2 slugs/cu ft C_M = 1.5

Solution

1. Virtual mass of sampler.

Volume of sampler	= $1/3\pi$ x3x3x5	= 47.12 cu ft
Mass of water in sampler	= 47.12x2	= 94.24 slugs
Added mass	= 1.5x47.12x2	= 141.36 slugs
Mass of sampler structure	= 320/32	= 10 slugs
Virtual mass	=	245.6 slugs

2. Inertia force.

Case a. The prudent operator brings the load to full speed in 8 seconds.

The average acceleration is then

$$\frac{8 \text{ ft/sec}}{8 \text{ sec}} = 1 \text{ ft/sec}^2$$

The average inertia force is then 245.6x1 = 245.6 lbs

The percent increase over the 200 lbs of static load due to the immersed weight of the sampler is then $\frac{245.5}{200}$ = 1.23 or 123% increase.

Case b. The "other" operator brings the load to full speed in 2 seconds.

The average acceleration is then

$$\frac{8 \text{ ft/sec}}{2 \text{ sec}} = 4 \text{ ft/sec}^2$$

The resulting inertia force is then 245.6x4 = 982 lbs

The percent increase is $\frac{(982)}{200}$ = 4.92 or 492%. This is almost five times the immersed weight of the sampler at rest.

3.5 All Forces Considered. Steady State Peak Tensions

When the hauling and lowering of equipment is done in a rough sea way the tension is no longer time independent, and inertia as well as drag forces must be considered.

To find, under these conditions, the tension in the cable at the shipboard end it is practical to first assume a zero

payout speed (winch secured). Assuming the travel path of the payload and the cable to be vertical (or nearly so) the dynamic tension $T(s,t)$ at the head sheave can then be evaluated with the help of Morisson's equation in one direction, namely

$$T(s,t) = W_P+W_L s+ \tfrac{1}{2} \rho(C_c\pi \dot{D}_c s + C_P A_P)\, V|V|+(m+m')\frac{dV}{dt} \quad (7.5)$$

where

s is the length of cable paid out

V is the vertical component of the head sheave speed at time t.

m is the mass of the cable and payload

m' is the added mass of the cable and payload

and $\frac{dV}{dt}$ is the vertical acceleration of the head sheave at time t.

Expression (7.5) implicitly stipulates that cable and payload rigidly follow the head sheave motion. In other words it treats the cable as a rigid bar. Despite this oversimplification expression (7.5) can be profitably used to calculate maxima of expected cable tension other than snap load.

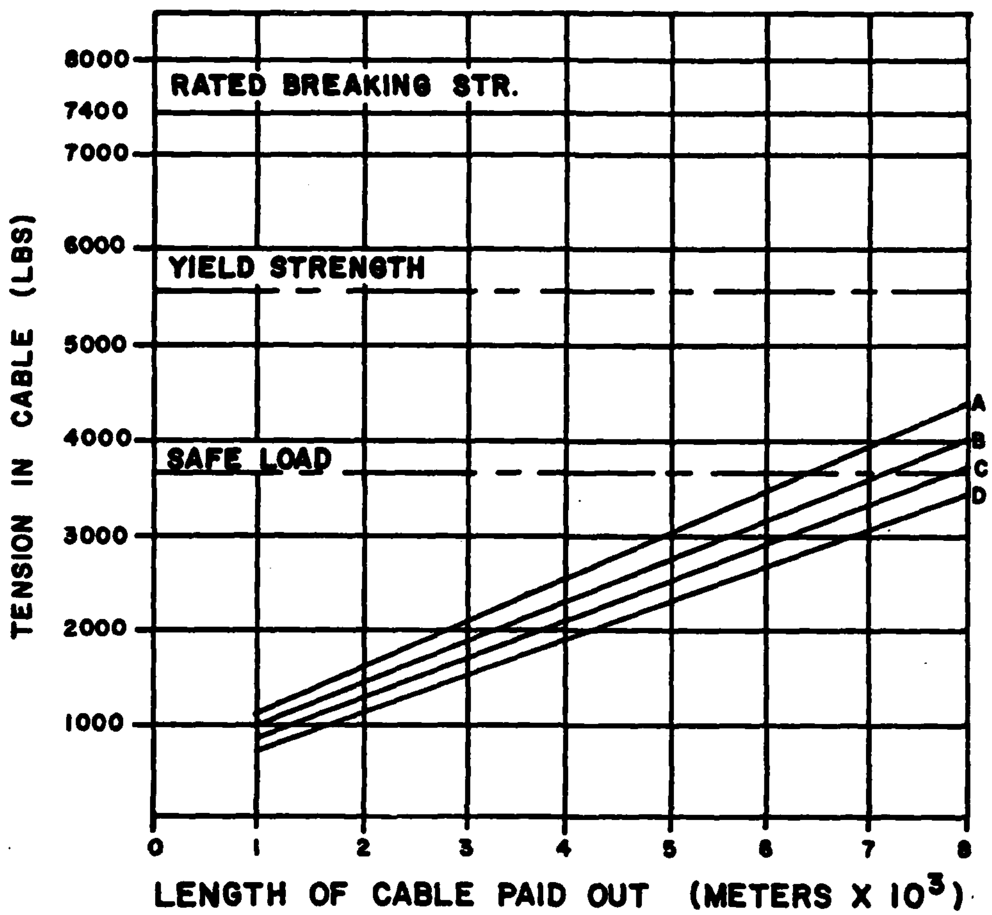
To this end one first derives the expression of head sheave vertical speed and acceleration as a function of ship geometry, wave amplitude and frequency. These expressions are then introduced in (7.5) and values of $T(s,t)$ are computed at discrete time intervals over a full wave period. The time of maximum dynamic tension occurrence can then be found by inspection.

The next step is to add the quasi-static contribution of drag due to hauling speed. A computation is made of the sheave velocity at time of maximum $T(s,t)$. The hauling speed is then added to this particular sheave velocity and the tension due to drag is then computed. Next the acceleration of the sheave is found for the time of maximum $T(s,t)$ and the corresponding inertia force is also computed.

The instantaneous maximum tension is then the sum of the total drag force, the inertia force, and the immersed cable and payload weight.

This simple computing procedure is best implemented with the help of a computer. Reference 5 presents in detail a derivation of head sheave speed and acceleration due to ship heave and roll and a program to evaluate peak tensions due to combined hauling and ship motion.

Once these calculations have been performed for a specific vessel the results obtained using this technique should be



SEA STATE 0
SHIP - ATLANTIS II
PERIOD = N.A.
HEAVE AMPLITUDE = 0.0
ANGLE OF ROLL = 0.0

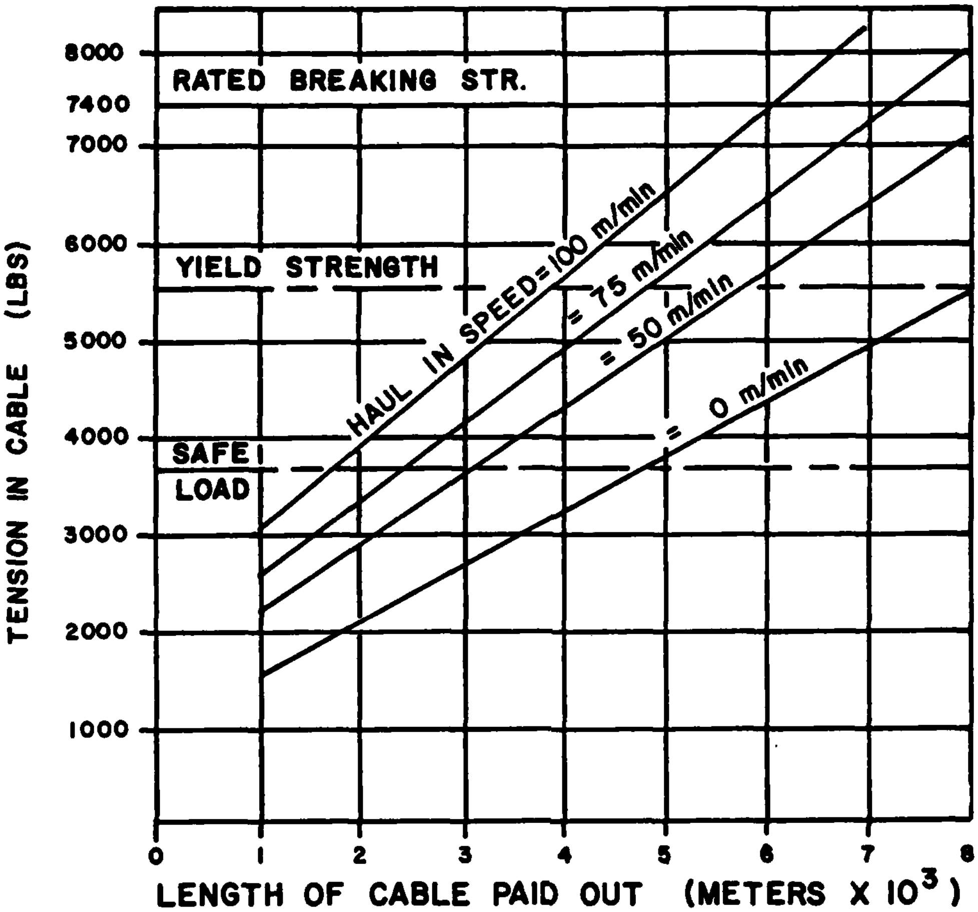
HAUL IN SPEED
A = 100 m/min
B = 75 m/min
C = 50 m/min
D = 0 m/min

CABLE CHARACTERISTICS =
WT/1000' = 145 lb.
DIAMETER = .303"
DRAG COEFF. = .01
RBS = 7400 lb.

INSTRUMENT CHARACTERISTICS =
IMMERSED WT. = 350 lb.
DRAG CONSTANT = 9.72 ft^2
VIRTUAL MASS = 21.0 SLUGS

PEAK TENSION AT HEAD SHEAVE VS.
LENGTH OF CABLE PAID OUT

FIGURE 7-2



SEA STATE 3
SHIP - ATLANTIS II
PERIOD = 8 SECONDS
HEAVE AMPLITUDE = 3 FT.
ANGLE OF ROLL = 15 DEGREES

CABLE CHARACTERISTICS =

WT/1000' = 145 lb.
DIAMETER = .303"
DRAG COEFF. = .01
RBS = 7400 lb.

INSTRUMENT CHARACTERISTICS =

IMMERSED WT. = 350 lb.
DRAG CONSTANT = 9.72 ft.^{2}
VIRTUAL MASS = 21.0 SLUGS

PEAK TENSION AT HEAD SHEAVE VS.
LENGTH OF CABLE PAID OUT

FIGURE 7-3

condensed and presented in a form easy to read. As an example, Figures 7.2 and 7.3 show the peak tensions calculated while hauling a CTD instrument package from the R/V ATLANTIS II under flat calm and sea state 3 conditions.

3.6 Snap Loads

A simple spring mass model (see Figure 7.4) can be used to predict the occurrence of snap loads and compute the ensuing cable tensions. In this model (Reference 2) the following assumptions are made:

- The motion of the payload is entirely vertical (one degree of freedom system).

- The mass of the cable is assumed to be a small fraction of the equipment mass. This would be the case for rather short lengths of cable (hundreds of meters instead of thousands), or if the cable is light (Kevlar line for example), or if the payload entrains a lot of water.

- The cable acts as a linear spring, the tension "T" being directly proportional to the cable elongation "Δ L", i.e.

$$T = k \ \Delta L \qquad (7.6)$$

In the elastic range of cable elongation the spring constant k is given by

$$k = \frac{EA}{L}$$

where E is the cable modulus of elasticity (psi)

A is the cable metallic area (sq.in)

L is the cable unstretched length (ft)

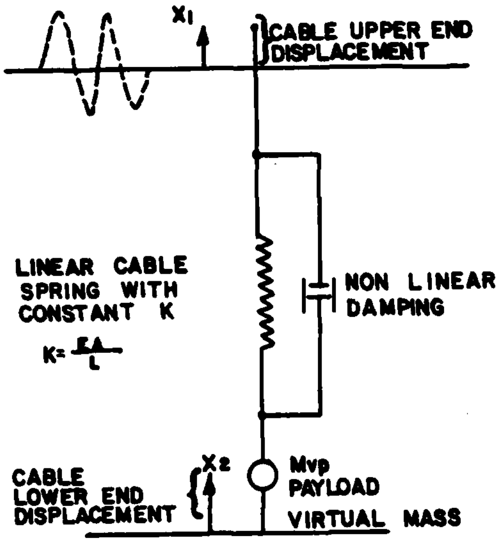
k will then be expressed in lbs/ft.

- Equipment drag is non linear and of the form

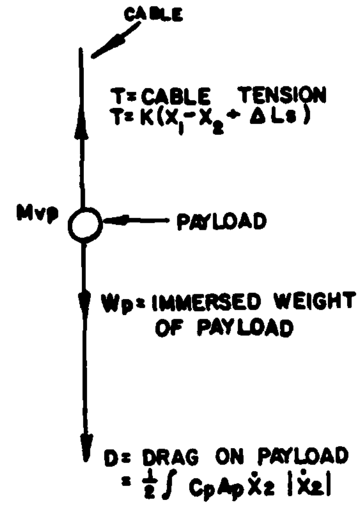
$$D = 1/2 \ C_p A_p V|V|$$

- Equipment drag is assumed much larger than the cable drag.

- The vertical displacement of the cable upper end can be described by an explicit function of time (such as a sinusoid).



CABLE/PAYLOAD SPRING MASS MODEL



PAYLOAD FREE BODY DIAGRAM

Figure 7-4

Applying Newton's law to the payload mass m_{v_p} (see Figure 7.4) yields the following equation of motion:

$$k(X_1 - X_2 + \Delta L_s) - W_p - \tfrac{1}{2} \rho C_p A_p \dot{X}_2 \, |\dot{X}_2| = m_{v_p} \ddot{X}_2$$

where

K	=	cable spring constant (lb/ft)
X_1	=	displacement of cable upper end (ft)
X_2	=	displacement of cable lower end (ft)
ΔL_S	=	cable elongation under pure static loading (ft)
W_p	=	immersed weight of payload (lbs)
m_{v_p}	=	virtual mass of payload (slugs)
$\dot{X}_2$	=	instantaneous speed of payload (ft/sec)
$\ddot{X}_2$	=	instantaneous acceleration of payload (ft/sec^2)

Noting that $k \, \Delta L_S = W_p$ this equation of motion reduces to

$$k(X_1 - X_2) - \tfrac{1}{2} \rho \, C_p A_p \, \dot{X}_2 \, |\dot{X}_2| = m_{v_p} \ddot{X}_2 \tag{7.7}$$

The instantaneous cable tension is then given by

$$T = W_p + k(X_1 - X_2) \tag{7.8}$$

The motion of the payload mass is governed by equation (7.7) as long as T=0.

If T, as given by (7.8) equals zero, then the payload is no longer pulled by the cable and a new equation of motion will prevail. Applying Newton's law to the payload in free flight yields

$$-W_p - \tfrac{1}{2} \rho C_p A_p X_2 \, |\dot{X}_2| = m_{v_p} \ddot{X}_2 \tag{7.9}$$

The system can be assumed to be initially at rest. At time t = 0 the upper end of the cable starts moving upwards. The ensuing motion of the payload is then found by integrating equation (7.7) using suitable numerical integration techniques. The author has found Euler's algorithm to be satisfactory provided the time increments are kept small.

Briefly stated, in this algorithm the acceleration of the payload over the time increment ΔT is given by:

$$\ddot{X}_2 = \text{Sum of the forces}/m_{v_p}$$

The speed is then simply

$$\dot{X}_2 = \dot{X}_{2(t-\Delta T)} + \ddot{X}_2 \, \Delta T$$

where $\dot{X}_{2(t-\Delta T)}$ is the speed at $t = t-\Delta T$

Similarly the displacement is then

$$X_2 = X_{2(t-\Delta T)} + \dot{X}_2 \, \Delta T$$

The tension is computed for each time interval, using (7.8). If it becomes positive then equation (7.7) prevails again. Speed and displacements when switching to a new equation are of course those computed in the time increment immediately preceeding the switch over.

Here again this computing procedure is best implemented with the help of a computer.

<u>Example 7.5</u>

To illustrate the use of this technique let us consider the response of a particular payload/cable system with characteristics as follows:

- Cable characteristics

  Type: = 3x19 wire rope

  Size = .375 inch

  Length = 3000 ft

  Immersed weight = 3000x .191 = 573 lbs

  Modulus of elasticity = 18,000,000 psi

  Metallic area = .1 sq. in

  Strength = 14,800 lbs

- Payload characteristics

  Type = Heavy instrument package

  Weight in air = 4200 lbs

  Weight of water displaced = 2200 lbs

  Added mass = 12 slugs

Normal area = 3.14 sq. ft

Drag coefficient = 1.0

- Input

The vertical displacement of the cable upper end is assumed to be given by $X_1 = 7 \sin \frac{2\pi}{4} t$

i.e. Displacement amplitude = 7 ft

Period = 4 secs

Solution

After transient, the response of the system - as calculated by a computer program implementing the technique just described - is as shown on Figure 7.5. From this figure one can see that the vertical displacement of the cable lower end varies from -3.5 ft to +14.8 ft whereas the upper end goes from -7 to +7 ft.

The peak tension obtained after the period of slack is 8100 lbs or four times as much as the static load (2000 lbs).

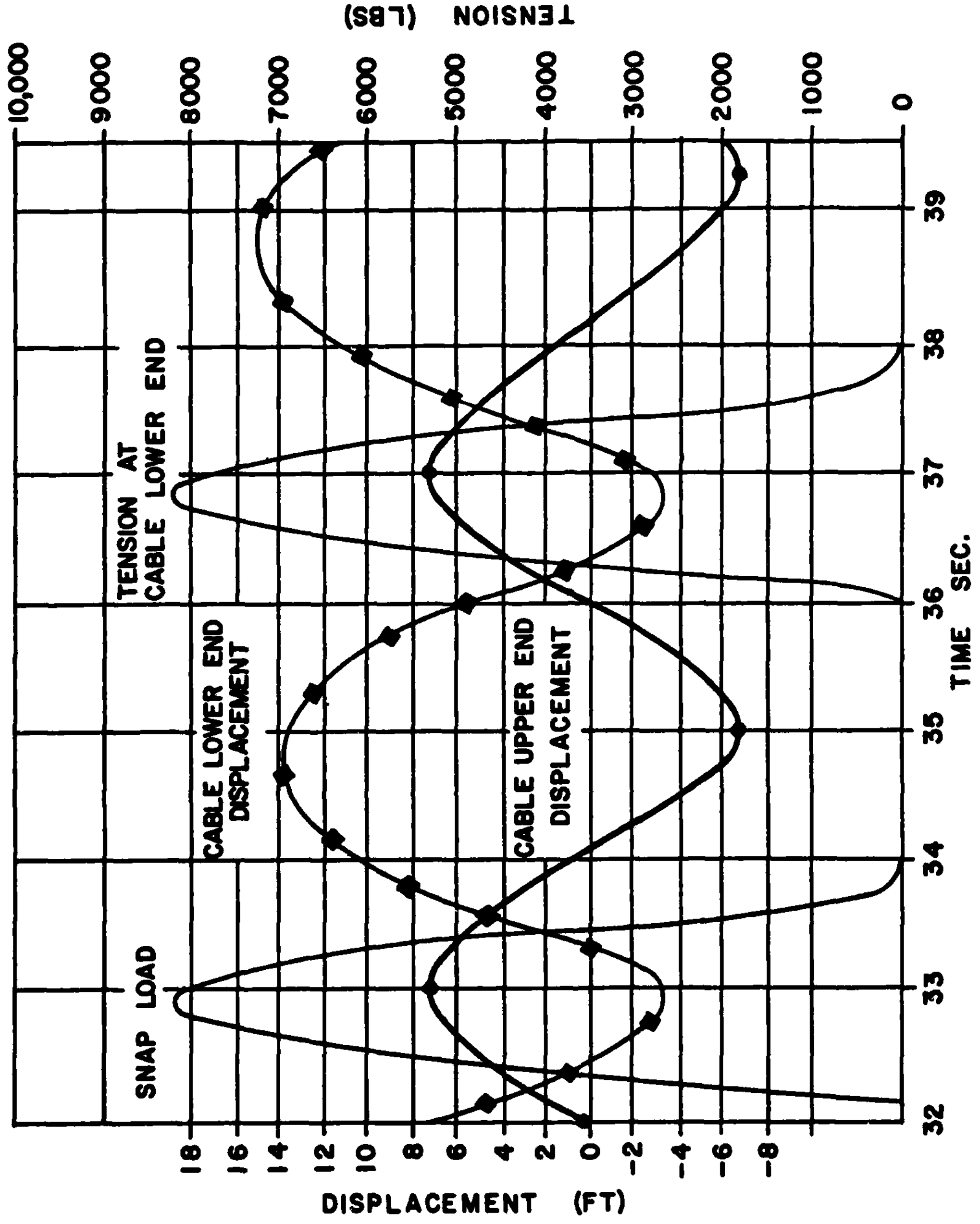
Under static load the comfortable cable safety factor is 14,800/2000 = 7.4.

Under snap load conditions this safety factor reduces to a mere 14,800/8100 = 1.83.

3.7 Advanced Cable Dynamics

The purpose of this section is to introduce the basic principles which govern cable dynamics. External forces acting on the cable - immersed weight, hydrodynamic drag, and inertia - have been reviewed. Formulas to calculate their magnitude have been given. How these forces interact to produce tension peaks and, equally important, slack conditions in the cable has been explained with the help of simple mathematical models. These introductory concepts will enable the reader to quantify the impact that payload weight and shape as well as hauling speed and ship motion may have on cable tension. They will help predict extreme conditions which after all are the most important ones.

Of course there is much more to the science of cable dynamics. Models treating the cable as a continuum in which deformation waves travel and dissipate have been proposed. Others treat the cable as a multiple degree of freedom system made of a number of point masses connected by linear and nonlinear springs and damping elements. Cable response to deterministic and random input has been investigated both in the time



CABLE LOWER END DISPLACEMENT AND TENSION AS A FUNCTION OF CABLE UPPER END DISPLACEMENT. (IMMERSED WEIGHT OF PAYLOAD EQUALS 2000 LBS)

FIGURE 7-5

and the frequency domain. Readers interested in this field are referred to the bibliography at the end of this chapter.

4.0 RECOMMENDATIONS

In this last section certain recommendations will be made to improve lowering mechanics and increase cable life expectancy.

4.1 Equipment Design Considerations

Equipment handled underwater is subjected to hydrodynamic forces not present when handled ashore. If the equipment has poor hydrodynamic characteristics it will impart high and unnecessary loads to the cable or cause it to kink and fail. To reduce or better yet suppress these detrimental conditions, underwater payloads should be designed with the following considerations in mind.

4.1.1 Stability. If the payload was permitted to free fall it should do so in a vertical path. In addition it should not tumble, flutter, or spin. A payload falling sideways, or kiting, will pull the cable at large angles from the vertical perhaps causing large cable bites and slack conditions. A tumbling or fluttering payload will jerk the cable at the point of attachment which may fail due to repeated bending. A spinning load can force hundreds of turns in the cable which will kink at the first opportunity.

To fall in a plumb, orderly way, the payload must be statically and dynamically stable. The payload is statically stable if it has a natural tendency to return to an upright steady state vertical flight. The payload is dynamically stable if it returns to its steady state upright vertical flight with oscillations of decaying amplitudes.

Investigating the static stability of a submerged object falling at some constant speed is straightforward and relatively easy. The step-by-step procedure involves:

- assume an initial tilt angle from the vertical
- resolve the drag forces into normal and tangential components
- compute the moments with respect to the body center of gravity induced by the buoyancy and drag forces
- sum the moments.

If the resultant moment tends to reduce the initial tilt angle then the object is statically stable at that angle. If not it will have a tendency to capsize. The process must then be repeated for increasing initial tilt angles.

If an object is found to be statically unstable over a large range of tilt angles (say from 0 to 45^{o}) then it is unfit for cable deployment. Its configuration must be altered until a proper combination of weight distribution and drag righting moment is found for all tilt angles considered. An example of such sensitivity study can be found in Reference 6.

Predicting the dynamic stability of free falling objects involves the simultaneous solution of six nonlinear partial differential equations. The mathematics required for this solution certainly go beyond the scope of this discussion. Interested readers should again consult the bibliography at the end of this chapter.

The following practical considerations if systematically implemented, can greatly improve the flight stability of cable lowered equipment and instrument packages.

- Weight Distribution. Packages should not be top heavy. Their center of gravity should be as low as possible and in all cases well below the center of buoyancy so as to provide a good righting moment.
- Shape. When placing instrumentation on a frame or equipment in some packaging form an effort should be made at reducing the top and bottom drag areas. A slender package will have much less drag as it travels vertically through the water than a fat, chubby one. Furthermore, it will entrain much less water and its added mass will be small.
- Symmetry. Vertical axisymmetry will greaty enhance flight stability. The payload weight should be distributed evenly around the vertical axis. If not the center of gravity will be off the center line and the package will not hang vertically from its point of cable attachment. The payload shape should also be axisymmetrical so that fluid induced forces cancel each other. As demonstated in tank tests, an acoustic pinger strapped on the outside of an instrument package frame causes the package to tilt and kite sideways as it sinks.
- Spin. In certain cases, equally distributed appendages can have the proper shape or inclination to induce a torque on the lowered equipment. This torque will force the equipment to spin. It is often possible to observe the spin of a load at the beginning (or the end) of its lowering. If detected, the condition causing the load to spin should be corrected.
- Control Surfaces. Control surfaces can sometimes be used to advantage to stablize an otherwise tumbling payload. Vertical fins located in the upper part of the package can provide a good righting moment. However, off center

loads equipped with vertical fins will steadily kite sideways. Furthermore, if one or several fins are bent, the load will spin. Horizontally mounted circular flaps are known to be very effective for stabilizig blunt cylinders. They are less sensitive to off-center loading. Their drawback is to reduce the cylinder terminal velocity.

4.1.2 Terminal Velocity. As previously explained to maintain tension in the lowering cable the speed of the payload fall must always exceed the speed of the cable. A payload with a small terminal velocity will therefore impose limits on the payout rate and/or the sea state in which the lowering operation can take place. If such operations are repetitive - as in the case of oceanographic profiling instrumentation - the ship time consumed in performing the lowering operation or in waiting for favorable weather becomes prohibitively expensive.

Achieving a reasonably fast terminal velocity should therefore be an important design consideration. The equipment designer should not hesitate at clamping some lead or steel blocks at the bottom of the payload to increase its weight. Doubling the instrument weight in most cases would have but a small effect on the lowering cable safety factor. Reducing the drag area and profiling the bottom of the equipment package will also increase the terminal velocity.

4.2 Equipment Handling Considerations

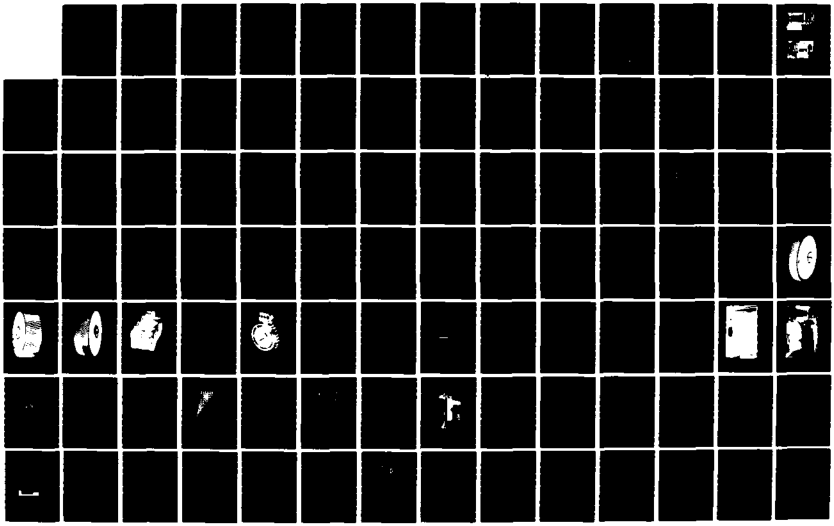
Now that the equipment has been properly shaped and trimmed, an investigation should be made of the operational limits necessary for its orderly and safe deployment.

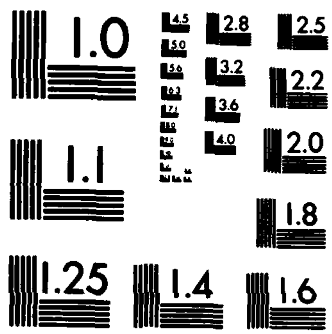
4.2.1 Depth Limits. Maximum tension occurs at the head sheave. As previously outlined this tension depends on the length of cable paid out, the weight and shape of the payload, the prevailing sea state and the hauling speed.

Whatever the actual condition of use may be this tension should not be permitted to exceed a value corresponding to a safety factor of two for most applications, and in no case larger than the yield strength of the cable (about 75% of cable breaking strength for most data logging cables). To help plan safe lowering, predictions of tension levels should therefore be readily available. If for example one had graphs of peak tension versus cable length for different hauling speeds and sea states of the type shown in Figures 7.2 and 7.3, then the maximum allowable cable length could be explicitly and rapidly established. In this case, the maximum length that the cable can (or should) have for a given sea state and a given hauling speed can be easily found from the intersection of the particular tension curve with the safe load (50% of RBS) line or the yield strength (75% of RBS) line.

AD-A132 513 HANDBOOK OF OCEANOGRAPHIC WINCH WIRE AND CABLE TECHNOLOGY(U) RHODE ISLANDUUNIV KINGSTON GRADUATE SCHOOL OF OCEANOGRAPHY A H DRISCOLL 18 OCT 82 4/5

UNCLASSIFIED N00014-82-G-0116 F/G 8/10 NL





MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

4.2.2 Winch Speeds Limits. Critical and/or repetitive lowering operations should certainly avoid slack cable conditions. Calculations of winch speeds which would cause the cable to become slack should be made as a function of sea state and length of cable paid out. These predictions should be made for every type of equipment lowered. They should be available in a convenient and tabular form. Limits on payout rates should then be set accordingly.

REFERENCES

1. E. A. Capadona, "Flexure Cycling Test of TRC7H4 Cable Acted on by Martin Decker Dynamometer," Preformed Line Products Company Test Report, January 1974.

2. J. E. Goeller and P. A. Laura, A Theoretical and Experimental Investigation of Impact Loads in Stranded Steel Cables During Longitudinal Excitation, Catholic University of America, Department of Mechanical Engineering, Themis Program, Report 70-2, 1970.

3. S. Hoerner, Fluid Dynamic Drag, Published by Author, Brick Town, New Jersey, 1965.

4. J. H. Pattison, et al, Handbook of Hydrodynamic Characteristics of Moored Array Components, David Taylor Naval Ship Research and Development Center, Report No. SPD-745.01, 1977.

5. H. O. Berteaux et al, A Study of CTD Cables and Lowering Systems, Woods Hole Oceanographic Institution, WHOI Reference 79-81, 1979.

6. M. Cook, Hydrodynamics of CTD Instrument Packages, Woods Hole Oceanographic Institution, WHOI Reference 81-76, 1981.

BIBLIOGRAPHY

Abkowitz, M. A., "Stability and Motion Control of Ocean Vehicles," The MIT Press, Cambridge, MA, 1969.

Albertson, N. D., "A Survey of Techniques for the Analysis and Design of Submerged Mooring Systems," U.S. Naval Civil Engineering Laboratory, Technical Report R-815, August 1974).

Berian, A. G., "Design and handling factors in the reliability and life of electrical wire lines," Proceedings of Interocean 1976, Dusseldorf, West Germany.

Berteaux, H. O., Buoy Engineering, John Wiley α Sons, New York, 1976.

Berteaux, H. O., Walden, R. G., Moller, D. A., Agrawal, Y. C., "A Study of CTD Cables and Lowering Systems," WHOI Technical Report 79-81, December 1979.

Booth, T. B., "Stability of Buoyant Underwater Vehicles, Part I, Predominantly Forward Motion," International Shipbuilding Progress, Vol. 24, No. 279, November 1977, pp 297-305.

Booth, T. B., "Stability of Buoyant Underwater Vehicles, Part II, Near Vertical Ascent," International Shipbuilding Progress, Vol. 24, No. 280, December 1977, pp 346-352.

Brainard, J. P., "Dynamic Analysis of a Single Point, Taut, Compound Mooring," Woods Hole Oceanographic Institution, WHOI Reference No. 71-42 (unpublished manuscript), Woods Hole, MA, June 1971.

Casarella, M. J., and J. I. Choo, "A Survey of Analytical Methods for Dynamic Simulation of Cable-Body Systems," Journal of Hydrodynamics, Vol. 7, No. 4, pp 137-144, October 1973.

Dessureault, J. G., "Batfish," A Depth Controllable Towed Body for Collecting Oceanographic Data, Ocean Engineering, Vol. 3, 1976, pp 99-111.

Dillon, D. B., "An Inventory of Current Mathematical Models of Scientific Data-Gathering Moors," Hydrospace-Challenger, Inc., Report TR 4450 0001, February 1973.

Dillon, D. B., "Verification of Computers of Cable System Dynamics," EGαG, WASCI, Technical Report for NCEL, 1981.

Doybe, G. R., Jr., Voracheik, J. J., "Investigation of Stability Characteristics of Tethered Balloon Systems," Goodyear Aerospace Corp., GER-15325, 30 July 1971.

Etkin, B., Dynamics of Flight - Stability and Control, John Wiley and Sons, New York, 1959.

Firebaugh, M. S., "An Analysis of the Dynamics of Towing Cables," Massachusetts Institute of Technology, Department of Ocean Engineering, Doctor of Science Thesis (unpublished manuscript), January 14, 1972.

Garrison, C. J., "Dynamic Response of Floating Bodies," OTC Paper 2067, May 1974.

Hoerner, S. F., "Fluid Dynamic Drag," 1965.

Hoerner, S. F., Borst, H. V., "Fluid Dynamic Lift," 1975.

Holmes, P., "Mechanics of Raising and Lowering Heavy Loads in the Deep Ocean: Cable and Payload Dynamics," U.S. Naval Civil Engineering Laboratory, Technical Report R-433, April 1977.

Hong, K., "Drag on Freely Falling Oceanographic Probes," Undersea Technology, November/December 1962.

Hong, S. T., "Frequency Domain Analysis for the Tension in a Taut Mooring Line," University of Washington, Department of Civil Engineering, Technical Report, No. SM 72-1, Seattle, Washington, July 1972.

Korvin, B. V. and Kronkovsky, Theory of Seakeeping, The Society of Naval Architects and Marine Engineers, New York, 1961.

Lamb, H., Hydrodynamics, 6th edition, Cambridge University Press, New york, 1932.

Liu, F. C., "Snap Loads in Lifting and Mooring Cable Systems induced by Surface Wave Conditions," Naval Civil Engineering Laboratory Technical Note N-1288, September 1973.

Marks, W., "The Application of Spectral Analysis and Statistics to Seakeeping," Technical & Research Bulletin No. 1-24, The Society of Naval Architects and Marine Engineers, New York, 1963.

Michel, W. H., "How to Calculate Wave Forces and Their Effects," Ocean Industry, May and June issues, 1967.

Migliore, H. and Zwibel, H., "Dynamic treatment of cable systems which change length with time," Canadian Conference of Applied Mechanics, Vancouver, B.S., May 1978.

Migliore, H. and R. L. Webster, "Current Methods for Analyzing Dynamic Cable Response," The Shock and Vibration Digest, Vol. II, No. 6, June 1979.

Myers, J. J., et al. Handbook of Ocean Engineering, McGraw-Hill, New York, 1969.

Nath, J. H., "Dynamics of Single Point Ocean Moorings of a Buoy and Numerical Model for Solution by Computer," Oregon State University, Department of Oceanography, Corvallis, Oregon, Reference 69-10, July 1969.

Newman, J. N., Marine Hydrodynamics, MIT Press, Cambridge, MA, 1977.

Pattison, J. H., "Hydrodynamic Drag of Some Candidate Surface Floats for Sonobuoy Applications," NSRDC Report 3735, August 1972.

Patton, K., "Tables of Hydrodynamic Mass Factors for Translational Motion," ASME Paper 65, WA/UNT-2, 1965.

Patton, K. T., "The Response of Cable Moored Axisymmetric Buoy in Ocean Wave Excitation," NUSC Technical Report 4331, Naval Underwater Systems Center, New London, Connecticut, June 1972.

Polachek, H., T. S. Walton, R. Meigia, and C. Dawson, "Transient Motion of an Elastic Cable Immersed in a Fluid," Mathematics of Computation, Vol. 17, No. 81, January 1963.

Shapiro, A. H., Shape and Flow, Anchor Books, Doubleday, Garden City, New York, 1961.

Skop, R. A., Ramberg, S. E. and Ferer, K. M., "Added Mass and Damping Forces on Circular Cylinders," NRL Report 7970, March 1976.

Vandiver, J. K., "Dynamic Analysis of a Launch and Recovery System for a Deep Submersible," Woods Hole Oceanographic Institution, Woods Hole, Massachusetts, WHOI Reference No. 69-88 (unpublished manuscript), December 1969.

Wang, H. T., "A Two-Degree-of-Freedom Model for the Two-Dimensional Dynamic Motions of Suspended Extensible Cable Systems," Department of the Navy, Naval Ship Research and Development Center, Bethesda, Maryland, Report 3663, October 1971.

Whicker, L. F., "Theoretical Analysis of the Effect of Ship Motion on Mooring Cables in Deep Water," David Taylor Model Basin Report 1221, Department of the Navy, Hydromechanics Laboratory Research and Develpment Report, March 1958.

Wilson, B. W., "Characteristics of Anchor Cables in Uniform Ocean Currents," A α M College of Texas, Department of Oceanography and Meteorology, Technical Report No. 201-1, April 1960.

Wendel, K., "Hydrodynamic Masses and Hydrodynamic Moments of Inertia," DTMB Translation No. 260, July 1956.

Yamamoto, T., Nath, J., and Slotta, L., "Wave Forces on Horizontal Submerged Cylinders," Oregon State University, Bulletin No. 47, April 1973.

Zarnick, E. E. and M. J. Casarella, "The Dynamics of a Ship Moored by a Cable System Under Sea State Excitation," The Catholic University of America, Institute of Ocean Science and Engineering, Washington, DC, Report 72-5, July 1972.

CHAPTER 8

Single Drum Winch Design

M. MARKEY

1.0 THEORY OF OPERATION 8-2
2.0 WINCH CAPACITIES 8-7
2.1 DRUM CHARTS 8-7
2.2 DRAG FORCES 8-11
2.3 SPECIAL APPLICATIONS 8-13
2.4 GEAR RANGES 8-13
3.0 WINCH POWER OPTIONS 8-15
3.1 HYDRAULIC WINCH DRIVES 8-16
3.2 ELECTRIC WINCH DRIVES 8-18
4.0 WINCH CONTROLS 8-19
5.0 GROOVED SHELLS AND FAIRLEAD DRIVES 8-20
6.0 INSTRUMENTATION 8-22
7.0 CONSTRUCTION DETAILS 8-24
8.0 ORDERING INFORMATION 8-26

1.0 THEORY OF OPERATION

The single drum research winch has been the mainstay of the oceanographic wire handling and storage business since the early 1930's and it relates to the wheel and the lamp bulb as one of the basic devices in the everyday experience of most of us. When discussing these machines, the oceanographic community applies an interesting mixture of English and Metric units when describing the operation of characterstics of the machine which might confuse the purist, but which are seldom a handicap. For example, wire diameters are usually given in inches, wire paid out is described about equally in both feet and meters. Pounds of force are used when referring to line pull and payload weights, but rarely kilograms; horsepower is the basic definition of power, and almost never kilowats, while line speeds are normally presented as meters per minute - although feet per minute remain common. It may well be the next generation before all the words and all the formulae have drifted into the more rational metric format. Winch manufacturers in general try to speak in any terminology that a customer brings to the table in dealing with his winch needs.

Research winches, of the single drum configuration, handle a range of wire diameters from 1/8 in. to 5/8 in., and scopes range from a few hundred feet to as much as 43,000 ft. capability (Figures 8-1 to 8-3). The weights of individual machines over this enormous range will vary from a few hundred pounds to the 35,000 lb. installed weight of the DUS-9 Type installed aboard R/V "KANA KEOKI."

The endless variety of operator's needs, and the equally endless variety in design solutions to those needs, is what keeps winch building an ever-changing profession and which creates a condition that mitigates against establishing the so-called "standard winch." There are, however, "cores of commonality" that exist for each winch manufacturer which assist in keeping individual unit costs within a realistic framework.

The single drum research winch possesses two fundamental advantages in its design which are as follows: 1) the wire path can be termed "simple," and 2) the machinery itself can be considered of "simple design." Once the wire has passed over the outboard sheave, the deck sheaves and any metering sheaves, it must only bend itself in a continuous direction around a drum core and around succeeding layers of itself. With the proper attention to the core diameter, the provision of starting grooves (LeBus shell, etc.), and the spooling tension, the bending of the wire into the drum can be a relatively easy process. Photographs 1 and 2 show the initial spooling of a 7/16 inch multi-coductor cable onto the type DUSH-8 (Figure 8-2), winch aboard R/V "ALPHA HELIX." Twenty-six layers of wire were spooled onto the drum with only a single fairlead head adjustment (on the 2nd layer). In watching the wire stay smooth and level, with perfectly square end reversals is a peak experience

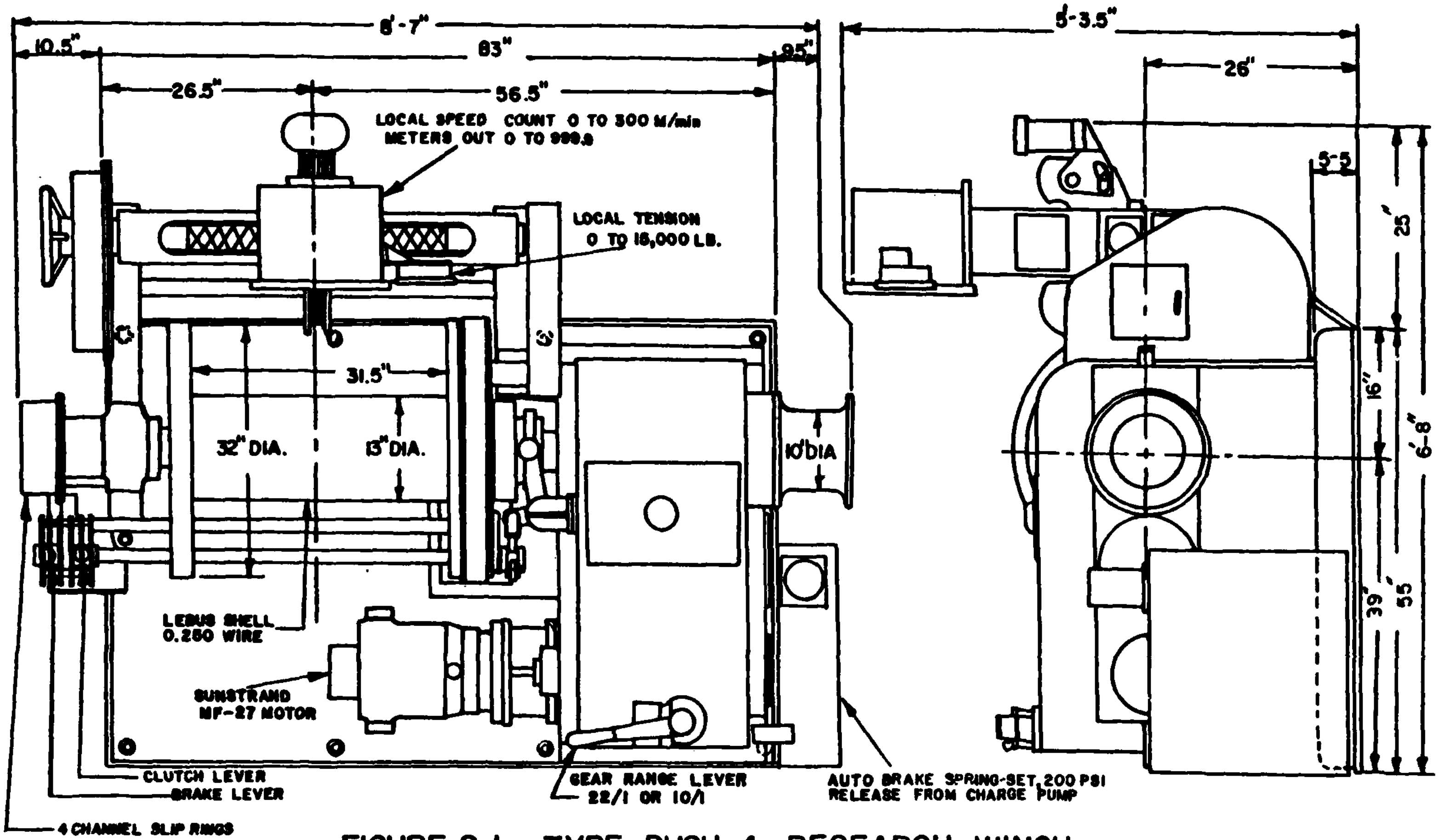


FIGURE 8-1 TYPE DUSH-4 RESEARCH WINCH

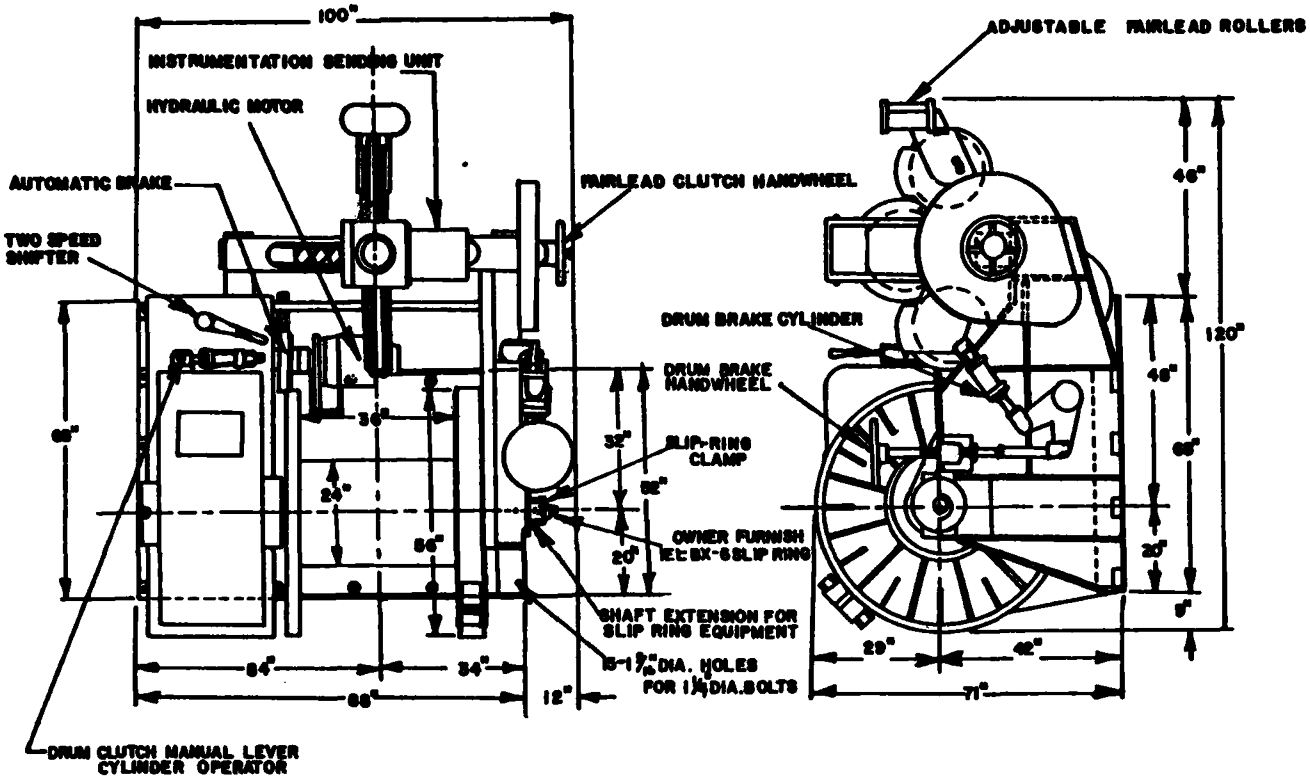


FIGURE 8-2 TYPE DUSH-8 HYDRAULIC RESEARCH WINCH

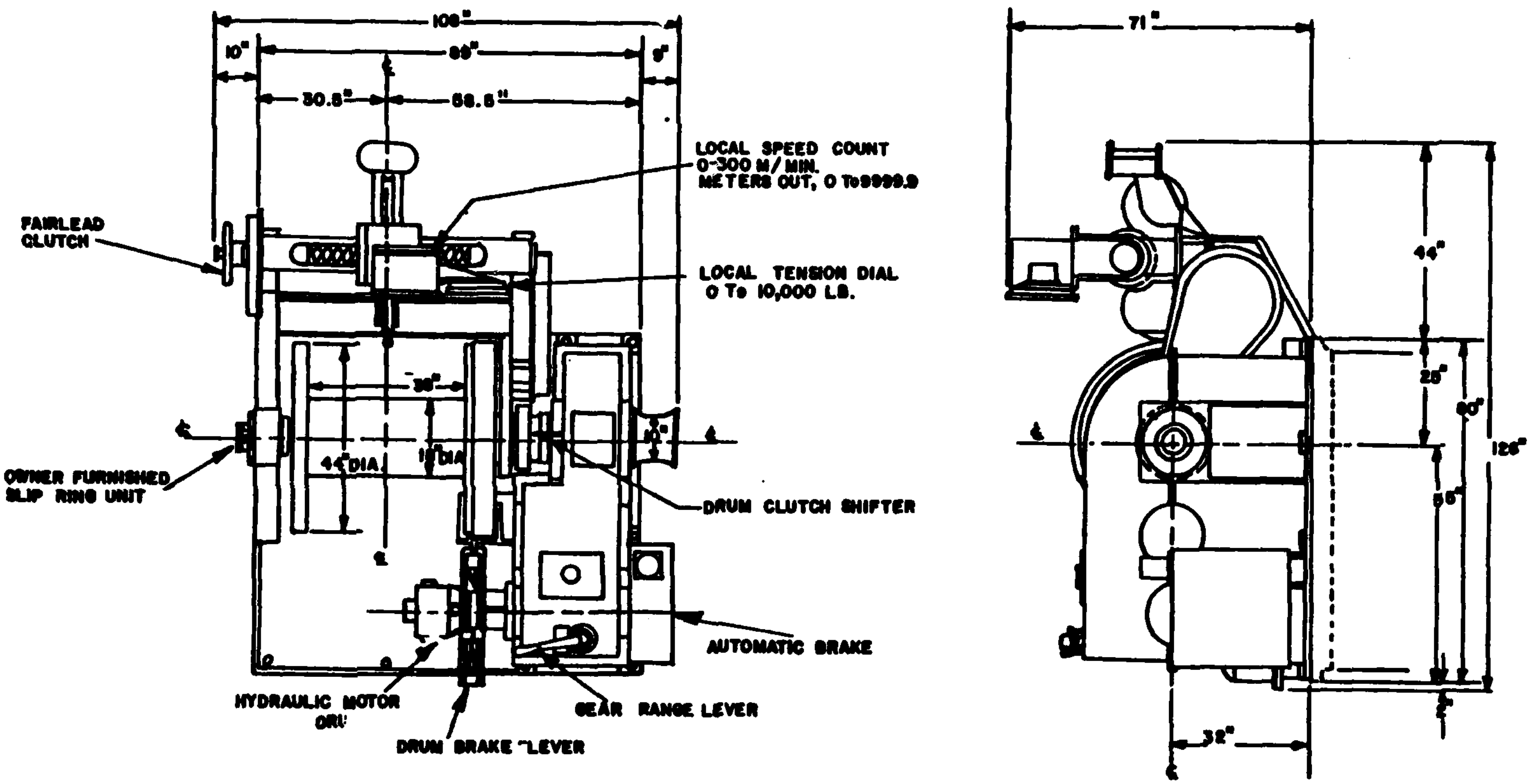
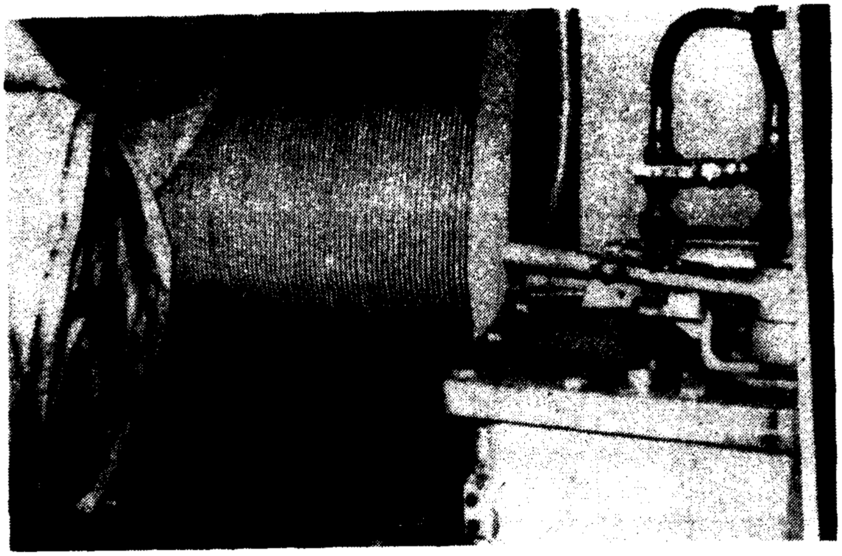
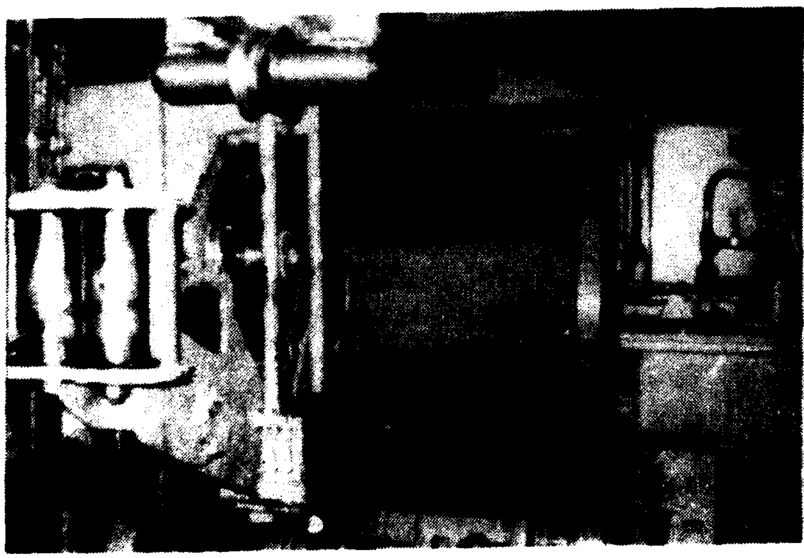


FIGURE 8-3 TYPE DUSH-5 RESEARCH WINCH





for any winch designer and the mark of a well-engineered winch system.

The single drum winch is, by definition, one piece of machinery which contains a single motor, gear train, storage drum, control set, and supporting structure. Other forms of wire handling equipment often require the addition of other components such as separate take-up reels and power units. While these added components are not necessarily a drawback, the designer of any winch system would do well to follow the working motto of a noted Seattle air frame manufactuer --- "KISS," or "Keep It Simple Stupid."

2.0 WINCH CAPACITIES

One of the fundamental characteristics of the single drum winch is the change in working radius that occurs as the winch drum fills and empties. Along with this change in working radius, the available line pull and line speed will also vary depending on the fullness of the drum. This section will deal with the implication of this characteristic and the means used to visualize the performance of individual winch systems.

2.1 Drum Charts

The drum chart format illustrates the current method of arriving at a basic set of winch performance specifications. This approach is as simple as equating the drum pulling capability at full scope to the nominal ELASTIC LIMIT of the wire size selected. There is little value in providing a pull rating greater than the capaciaty of the wire to survive; however, if several wire sizes are to be used on the same winch, the machine should be rated to handle the largest of them to its elastic limit. The alternative concept of rating the winch to the breaking strength of the largest wire in use may not be wholly valid due to the overload capacity inherent in single drum winch design, which will usually allow the winch to meet or exceed the rated strength of the wire. Special circumstances, such as user requirements, etc., may dictate the rating of the winch to a higher capacity. Also, in applications where payload speed is the critical factor, a lesser pull rating in order to achieve desired speed may be acceptable to the user as a trade-off.

Table 1 represents a "drum chart" for the R/V "ALPHA HELIX" DUSH-8 winch, with 6,000 meters of 7/16 in. wire as installed. The left hand block of information describes the geometry of the wire volume, and the accumulation of wire payed overboard, and the weight accumulation of that wire. The middle and right hand blocks provide winch pulls and line speeds in the two gear ranges built into the winch.

The data becomes interesting when the "Pull in Excess of Wire Weight" columns is considered. A "shallow" minimum

value shows up - at the 20th layer of wire where, with nearly 6,200 ft. (1,890 m) payed out, there is a 9,225 pound (4,193kg) value of pull assignable to the sum of the payload weight, the payload drag, the wire drag, and any acceleration forces which may be involved. It is this data column that tells the operator what he can do with his machine, with regards to load handling capacity.

The implication here is that the operator could pick a 9,467 lb. (4,303kg) load off the deck and lower it to any depth below 6,200 ft., but theoretically be unable to bring it back aboard. In this case, such a small difference would vanish into the design margins, drive efficiencies and overloads inherent in the winch system and the 9,467 lb. would surely come home. However, it might not come back at rated speeds, since drag forces on the payload do exist and must be overcome. By reducing the winch speed, the operator also reduces drag forces acting on the payload and a theoretically marginal load should come up.

Also in Table 1, it will be noted that the right hand data block, showing the High Speed Gear Range reflects a difference in gear ratio of only 1.7 to 1, but the effect on pulling capacity is much more than the linear effect upon line speed. In this case the weight of the wire becomes the larger percentage of the available torque and the entire winch capacity picture changes dramatically. Here the minimum "Pull In Excess" capacity is 4,128 lb. (1,917kg) versus the full drum lift capacity of 5,316 lb., (2,416kg). The operator, under these conditions, may not be able to recover a marginal load without stopping to shift back into the high pull lower speed range.

Table 1 indicates that the 6,000 meter length of wire actually installed on the unit occupied only 26 layers of a 35 layer total drum volume. This condition reflects a not unusual element in the specification and design process and represents a step in the proper direction to maximize winch usage relative to changing user demands.

As a second example of the usefulness of drum charts, Table 2 displays the affect of installing 26,240 ft. (8,000 m) of 7/16 in. wire on the same DUSH-8 winch. The greater weight of wire overboard is shown in the "Pull In Excess of Wire Weight" columns where it is shown that in the high pull range, the "minimum" pull-in-excess load has dropped from 9,225 lb. (4,193kg) to 7,552 lb. (3,433kg) and in the high speed range the "minimum" pull-in-excess load has dropped from 4,128 lb. (1,876kg) to 2,455 lb. (1,116kg). Even if the same length of wire were paid out in both examples (Tables 1 and 2), the drum radius difference generated by a greater wire length on the same winch would exert a dramatic difference on the winch's pulling capacity. For example, starting at the 26th layer on the drum with a total available wire length of 6,000 m., a 14,000 ft. (4,267 m), cast would result in a 10,042 lb. (4,565kg) pull in

* indicates MINIMUM value of pull available for Payload, Drag, & Acceleration
** indicates mid-scope layer of wire

					High Pull Gear Range 58.428/1			High Speed Gear Range 34.668/1			
Layer No.	Inch, Pitch Diam.	Feet of Wire Per Wrap	Feet of Wire Payed Out	Pounds... Weight of Wire in H_2O	Total Drum Pull Capacity lb.	Drum Pull in Excess of Wire Weight	Line Speed ft/min	Total Drum Pull Capacity	Drum Pull in Excess of Wire Weight	Line Speed ft/min	Layer No.
35	54.032	14.14	("full drum" would spool 30,009 ft of 0.428 in. cable)								
34											
33											
32											
31											
30	49.752	13.02	Starting With 19,680 feet								
29											
28											
27											
26	46.328	12.13	570	145	9,612	9,467	218	5,461	5,316	385	26
25	45.472	11.90	1,557	397	9,793	9,396	214	5,554	5,167	378	25
24											24
23											23
22											22
21						9,233					21
20	41.192	10.78	6,216	1,585	10,810	*9,225	194	6,142	4,557	342	20
19						9,232					19
18											18
17											17
16											16
15	36.912	9.66	**10,410	2,654	12,064	9,410	174	6,854	4,200	307	15
14											14
13									4,139		13
12									*4,128		12
11									4,130		11
10	32.632	8.54	14,139	3,605	13,647	10,042	154	7,753	4,148	271	10
9											9
8											8
7											7
6											6
5	28.352	7.42	17,403	4,437	15,707	11,270	133	8,924	4,487	235	5
4											4
3											3
2											2
1	24.928	6.53	19,680	5,018	17,864	12,846	117	10,150	5,132	207	1

MARKEY MACHINERY COMPANY CAPACITY CHART - MMCo. Type DUSH-8 Research Winch
Serial No. 13127

Based on spooling 6,000 meters (19,680 ft.) of 0.428" diam. E.M. Blue multi-conductor cable with in-water weight of .255#/ft

TABLE I

("full drum" would spool 30,009 ft of 0.428 in. cable)

* indicates MINIMUM vlaue of pull available for Payload, Drag, & Acceleration
** indicates mid-scope layer of wire

Layer No.	Inch, Pitch Diam.	Feet of Wire Per Wrap	Feet of Wire Per Layer	Feet of Wire Payed Out (Starting with 8,000 Meters)	Weight of Wire in Water Pounds	High Pull Gear Range 58.428/1: Total Drum Pull.. Pound Capacity	High Pull Gear Range 58.428/1: Drum Pull in Excess of Wire Weight	High Pull Gear Range 58.428/1: Line Speed ft/min	High Speed Gear Range 34.668/1: Total Drum Pull.. Pound Capacity	High Speed Gear Range 34.668/1: Drum Pull in Excess of Wire Weight	High Speed Gear Range 34.668/1: Line Speed ft/min	Layer No.
35	54.03	14.14										
34												
33												
32	51.46	13.47	1,118	814	207	8,653	8,446	242	4,916	2,709	427	32
31												31
30	49.75	13.02	1,080	2,993	763	8,951	8,188	234	5,086	4,323	413	30
29												29
28												28
27												27
26												26
25	45.57	11.90	987	8,117	2,070	9,793	7,723	214	5,564	3,494	376	25
24												24
23												23
22												22
21												21
20	41.19	10.78	894	12,776	3,528	10,810	* 7,552	194	6,142	2,884	342	20
19												19
** 18	39.48	10.34	858	14,510	3,700	11,279	7,579	186	6,408	2,708	328	18
17												17
16												16
15	36.91	9.66	801	16,970	4,327	12,064	7,737	174	6,854	2,527	307	15
14												14
13												13
12									7,367	* 2,455	285	12
11												11
10	32.63	8.54	708	20,699	5,278	13,647	8,369	154	7,753	2,475	271	10
9												9
8												8
7												7
6												6
5	28.35	7.42	615	23,963	6,110	15,707	9,597	133	8,924	2,814	235	5
4												4
3												3
2												2
1	24.93	6.53	524	26,240	6,691	17,864	11,173	117	10,150	3,459	207	1

MARKEY MACHINERY COMPANY CAPACITY CHART - MMCo. Type DUSH-8 Research Winch
Serial No. 13127

Based on spooling 8,000 meters (26,240 ft.) of 0.428" diam. E.M. Blue multi-conductor cable with in-water weight of .255 #/ft

TABLE 2

excess of wire weight. With the same depth cast, but with an 8,000 m available wire length on the winch drum, the user is now operating at the 32nd layer with only a 7,500 lb. (3,409kg) pull in excess of wire weight avialable.

The drum charts shown in Tables 1 and 2 have effectivley given the winch user a detailed set of operating conditions that can be expected for various wire lengths. With increasing demands being placed on winch systems by the everchanging nature of ocean science, the drum chart can be used effectively to predict the potential of a given winch under varying conditions of wire length, size, payload weight, and line speed. Alternatively, the drum chart approach provides an analytical tool which can be effectivley employed by the user to evaluate a winch system when non-conventional wire sizes and experiments are planned.

2.2 Drag Forces

Some empirical work was done in the area of attempting to assign values for payloads and wire drag in order to balance the winch design against specific payload requirements. Table 3 shows a Drum Chart for the 1/4 in. Type DUSH-4 installed aboard R/V "NEW HORIZON" with two arbitrary values assigned to the total drag forces and the payload. The object was to balance the drum's torque against the assumed torque requirements. This approach, although well intentioned, was seen as reaching beyond the available data, and into the realm of conjecture.

The several winches manufactured via this procedure have performed well in the field and it is relatively safe to assume that the initial estimates of drag forces were reasonably accurate. However, until accurate drag values can be provided by the user community, the use of drag forces as legitimate deductables, along with wire weights, etc., cannot be performed with any degree of precision. Until that time, drum design torque will be calculated based on the elastic limit of the largest wire in use for each winch.

The point here is that marginal load situations can be experienced as a result of payload and wire drag even though the simple weight calculation will show an acceptable condition. With the development of new instrumentation and heavier demands on the winch systems, the intangibles of payload drag will increasingly influence the performance of the winches in use today. Calculations of the drag forces involved in new equipment should be made and the resulting values factored into the drum chart so that the winch performance can be assessed under these new circumstances.

				High PULL					Shift Ratio 2.2/1	High SPEED			
Layer	Inch, Pitch Diam.	Feet of Wire Payed Out	Pounds of Wire Payed Out H_2O-Wt.	Assumed Nominal DRAG of Payload & Wire Pounds	Pounds Payload H_2O-Wt.	Rated Pounds Total Line Tension	Pounds Max Line Tension 1600 psi Drive	Ft/Min	Assumed Nominal DRAG of Payload & Wire Pounds	Pounds Payload H_2O-Wt.	Rated Pounds Total Line Tension	Pounds Max Line Tension 1600 psi Drive	Ft/Min Line Speed
35	30.25	1,000	120	350	4,500	4,970	5,125	390	700	500	1,320	2,330	860
34													
33													
32													
31													
30	27.75	5,723	687	350	4,500	5,537	5,590	360	700	500	1,887	2,540	790
29													
28													
27													
26													
25	25.25	10,031	1,204	400	4,500	6,104	6,145	327	800	500	2,504	2,790	715
24													
23													
22													
21	23.25	13,184	1,582	400	4,500	6,482	6,670	300	800	500	2,882	3,030	660
20	22.75	13,931	1,672	450	4,500	6,622	6,820	295	900	500	3,072	3,100	645
19													
18													
17													
16													
15	20.25	17,418	2,090	450	4,500	7,040	7,660	260	900	500	3,490	3,490	575
14													
13													
12													
11													
10	17.75	20,496	2,459	500	4,500	7,459	8,740	225	1,000	500	3,959	3,970	505
9													
8													
7													
6													
5	15.25	23,162	2,779	500	4,500	7,779	10,170	195	1,000	500	4,279	4,620	430
4													
3													
2													
1	13.25	25,000	3,000	500	4,500	8,000	11,700	170	1,000	500	4,500	5,320	375

MARKEY MACHINERY COMPANY CAPACITY CHART - MMCo. Type DUSH-4 Hydraulic Research Winch
Serial No. 61577

Based on full drum with 25,000 ft. of 0.250" diam. wire, with assumed water-weight of 12 lb./100 ft.

TABLE 3

2.3 Special Applications

One final use of the drum chart approach to winch design is reflected in Table 4 which rates a proposed DESH-6, shallow water research winch capable of handling 1,000 ft. (305 m) of 3/8 in. wire rope. Under these operating conditions the weight of wire is nearly negligible with the Pull Capacity column becoming, in effect, the only data required by the operator. Even though this application is a rather simple one, the design approach of pulling power matched to the wires elastic limit and the use of dual gear ranges in the winch is still a valid procedure.

2.4 Gear Ranges

The drum charts have all shown dual gear ranges and this feature has been a standard in Markey Machinery Company research winches for many years. While most working loads do not approach the maximum capability of the wire, the operators tend to need all the payout and recovery speed that is possible in order to minimize station time since most installations do not have as much horsepower available as the builder and operators would like. The dual gear range has been used to partially overcome this problem.

The fast range is designed for use in most payout situations, as well as in the majority of recovery situations experienced at sea.

The selection of ratio differences between the two speed ranges is crucial to the maximum performance of the winch and is governed by a balancing of line speed versus pulling power. A look at the various drum charts in this section will show how the Pull In Excess of Wire Weight drops as the gear ratio is reduced in order to achieve more line speed. If this trade-off is carried too far, the winch can conceivably struggle to recover the wire itself exclusive of any payload weight or drag.

Since the drag forces acting on the wire and its payload increase with the speed at which it is pulled up through the water column, the available line pull is further reduced as the speed increases. During the design of a winch system, it is important that the user and the manufacaturers communicate effectively as to the requirements of the user and the trade-offs he will have to make in order to meet those requirements. To date, a shift range of between 1.5:1 and 3:1 have been found to be acceptable for oceanographic applications.

Line speed in a winch system tends to be one of those items which is always considered to be in short supply from the users standpoint. It must be remembered that a winch's hoisting speed is limited by the available horsepower and major speed increases are virtually impossible to obtain without expensive

* * Mid-scope layer of wire, with "average" speeds of 25 M/min.and 75 M/min.

Layer Number	1	2	3	**4	5	6	7	8	9	10	
Pitch Diameter inches	18.375	19.125	19.875	20.625	21.375	22.125	22.875	23.625	24.375	25.125	
Feet per Layer	144	150	156	162	168	174	180	185	191	197	
Cumulative Feet Payed Out	1,000	856	706	550	388	220	46				
Weight of Wire Out - Pounds	191	163	135	105	74	42	9				
Winch Pull Capacity - pounds	11,000	10,568	10,170	9,800	9,456	9,135	8,836				High PULL Range
Line Speed - feet/minute	73	76	79	82	85	88	91				
Winch Pull Capacity - pounds	3,667	3,522	3,390	3,266	3,152	3,045					High SPEED Range
Line Speed - feet/minute	218	227	236	246	254	263	273				
Layer Number	1	2	3	**4	5	6	7				

MARKEY MACHINERY COMPANY CAPACITY CHART -
MMCO. Type DESH-6 Research Winch

Based upon 1,000 ft. of 0.375" diam. 3 x 19 wire rope, with 11,000 lb. elastic limit. In-water weight approx. 0.191 lb./ft.

Drum dimensions 18" diam. barrel x 12" face width x 26" diam. flanges Dual-range winch gearing, with approx. 3/1 ratio shift

TABLE 4

winch repowering. As mentioned earlier, the available or assignable horsepower of a given winch system dictates its ultimate performance.

3.0 WINCH POWER OPTIONS

The design of an adequately powered winch drive involves an understanding and mutual agreement by all parties concerning the line pulls and line speeds which will be required at sea. The selection of motors, controls and energy sources involves the designer's experience, and his best judgements regarding the mechanical efficiency of the machine and the losses within the winch drive.

The horsepower required to turn the winch at some desired speed and line pull can be easily derived from the following formula where: the horsepower coming out of a winch drum is given by the formula -

$$\text{Horsepower} = \frac{\text{Line Pull (lb.)} \times \text{Line Speed (ft/min)}}{33{,}000}$$

In a spur geared winch with all shafts carried on anti-friction bearings, a mechanical efficiency of 80% to 85% can be expected. This combined with the derived horsepower, using the above formula, will bring the winch designer up to the point where a power input to the winch can be determined.

Basically there are two types of winch power units available today, either electric or hydraulic drives, or a combination of the two. In some instances direct drive diesel or diesel hydraulic units have been employed. However, for the purposes of this section we will concentrate on the first two power options.

With an electrically driven winch, the input derived from the formula gives the rating for the winch motor which is then rounded up to the next standard nameplate rating available. There exists a concept regarding electric winches whereby they are spoken of by their winch motor ratings. For instance, aboard the R/V "WECOMA," the three Markey Machinery Company machines are referred to as the 25 H.P. unit, the 40 H.P. unit, and the 125 H.P. unit with little consideration being given to the prime horsepower necessary to power the equipment involved.

When considering hydraulically driven winches, however, it is natural to speak in terms of the prime power, or the power required to drive the hydraulic pump. Hydraulic winches carry this "P.R." burden in addition to the actual inefficiencies of the hydraulic transmission. As an example, assume a winch which produced a mid-scope drum output of 5,000 lb. line pull, at a line speed of 200 ft./min. This set of conditions produces a "drum horsepower" of 30 H.P. and as an electric winch the system

would be driven by a 40 H.P. motor and it would likely be called a "40 H.P. winch." If its hydraulic alternative were fitted with a simple vane type system, the 30 H.P. winch output would have to be multiplied by approximatley 2.4/1 resulting in pump drive requirement of 72 H.P. One would then specify a 75 H.P. electric motor, or a 75 to 80 H.P. PTO clutch to power the winch. If the 30 H.P. winch output were provided with a more efficient piston type hydraulic system, the winch output horsepower would be multiplied by approximately 2.2 to 1, requiring 66 H.P. to the pump. The choice here would be to either overload a 60 H.P. electric motor, or move up to the same 75 H.P.

Hydraulics manufacturers and some winch builders will argue about this seemingly high power multiplier between the winch drum and the pump drive. They will point to Motor and pump efficiency curves and note that the produce of those efficiencies is in "no way" as low as experience indicates. The point is that a hydraulic loop is made up of valves, elbows, filters, screens, as well as pumps and motors which represent frictional losses within the system that reduces the power units output horsepower. These losses are reflected in an old myth which was used to describe a hydralic system. In effect, it states that, "Nine elbows equals one plug." While the statement is not literally true, it does point out that hydraulic system losses represent significant numbers.

3.1 Hydraulic Winch Drives

Hydraulic power units make up the winch drive system most often installed on present-day winches. The initial cost of this type of system should be less than an equivalent electrical system, however, this initial advantage may lessen when installation and start-up costs are considered.

Several Markey Machinery Company winches have done quite well when driven with simple vane type motors with fixed-flow vane pumps. The usual approach in this case is to apply two vane motors to the winch and to install a six-port valve which allows the motors to be driven either in SERIES or in PARALLEL. The parallel motors divide the flow and provide "half speed" while putting the full pressure drop across the two motors for maximum torque. With the two motors connected in series, both motors see the full hydraulic flow and produce a "full speed" drive with the hydraulic pressure cascading through the two motors, resulting in a half torque output. In this application the two operating speed pull ratings are provided with a "constant horsepower" characteristic, and when combined with a manual reversing valve with excellent throttling control, the operator obtains considerable winch flexibility at a minimum cost.

This theme can be expanded by the use of multiple pumps with unloading valves staging in various flow totals, however, an unpredictable speed change could come at inopportune moments.

Dual range gearing is often used with the two motors to provide four working ratings for the winch which is perhaps enough for most situations.

The majority of hydraulic specifications center on the variable displacement piston pump and the fixed displacement piston motor with the useful intent of providing a truly variable speed control. Most variable pumps are below deck where a remote control system for the stroke is needed. Before commenting on the remote pump controls, a word is needed about the REVERSING aspect of the hdyrostatic loop circuit.

Our experience is that the OFF status of most piston pumps is a delicate one -- truly zero flow is needed to keep a winch still, and creep in either direction is a nuisance at best. Remote controls add to the need for stroker precision and adjustment, and it has become our standard to sidestep the zero-stroke problem. The introduction of a large reversing valve into the loop makes certain that the winch motor sees zero flow when the valve is in the "Pump-to-EXHAUST" position, regardless of the pump stroke setting. Taking this a step further, Markey makes and installs positive "zero-stroke sleeves" into the servo pistons of the piston pumps so that there is no way for the pump to accidentally route flow to the "other direction." Two advantages result from this technique: 1) The stroke sleeves establish a high-pressure side and a low-pressure side of the loop with potential advantages in piping cost, filter locations, etc. 2) The positive reversing valve can also be considered as a "kill switch" which supplements the usual stopping procedure of reducing the pump stroke gradually to zero.

The choice between "closed-loop" hydrostatics and "open-loop" hydrostatics appears to be one of convenience as regards the charge pump. The 200 psi auxiliary charge pump on most closed-loop piston pumps is useful not only for the pumps own control servos, but also for the operation of automatic winch brakes and in some cases, hydraulic types of remote controls. It should be noted that the charge pump is critical to the operation of the main pump, and outside uses of its flow should be minimized. If the charge circuit pressure drops from its healthy 200 psi (on a typical Sundstrand) too much below 150 psi, the main pump comes off stroke by itself and the system goes soggy. A good gage should always be installed in the charge pump pressure tap, since a falling of the pressure is an excellent indication that the big pump has swallwed something unpalatable, scored itself, and will shortly require attention.

System longevity is enhanced by running the pump at 1,200 rpm rather than 1,800 rpm, and 2500 psi normal operating pressure is a more kindly cut-off pressure than 3,500 psi. As always a design balance must be struck between reduced initial investment with smaller component size against longer component life. Given the level of difficulty involved in major on-board hydraulic service, it is easy to recommend the lower stress

system parameters.

Winch drive horsepower can be applied either in the luxurious "corner horsepower" quantity, or the cheaper, but less predictable "horsepower limited" amount. The drum charts shown in Table 8-1 to 8-4 are based upon providing enough pump drive power to take the system to maximum flow rate at the maximum pressure "corner" of the pump performance diagram. This gives the operator controlled access to the needed line pull at any selected line speed. However, if pump drive power is in short supply, the best of both the PULL and the SPEED worlds can still be furnished, but not simultaneously. If the pump is fitted with a "horsepower limiter" control, it will respond to the operator's request for flow as long as the load does not demand too much pressure. If the load becomes too heavy, this will create a pressure increase in the loop, and the limiter control will take over to reduce the stroke -- regardless of where the control cab handle is placed. The system can be run with a smaller electric pump drive or PTO clutch, but the operator is never quite certain of the speed at which things will happen.

We should say a word about the relative uselessness of considering hydraulic components in terms of their horsepower ratings. In producing a winch design, the horsepower output of the hydraulic motor simply does not enter the picture. One thinks in two separated parameters -- torque and speed. Naturally their product results in horsepower, but it is only in separate inch-pounds and in rpm values that meaningful results occur. Taking this a step further, the same mental separation applies to the entire circuit.

FLOW RATE creates WINCH SPEED

PRESSURE DIFFERENTIAL across the motor creates WINCH PULL

3.2 Electric Winch Drives

Electrical drives for research winches have been somewhat eclipsed recently, but we consider that they retain virtues well worth considering. Ruling out the pushbutton squirrel cage type drive as being single speed or two-speed, with abrupt starts and stops, we move to the variable speed drives. Multi-speed "wound rotor" AC drives have the problem of always running to the wide-open synchronous speed under light load conditions. The several DC systems have had the widest application.

Constant-voltage DC drives of the simple five speed type have been applied and have done good work, but when the partial speeds are provided by routing a major portion of the current through grids, there is much heat created and wasted when long periods must be spent in the lower speed points.

Variable-Voltage DC drives are either of the "rotating source" type, with a motor-generator or diesel-generator set providing the DC, or more recently, with SCR power units. Excellent results can be produced with generator set power, since field weakening can be utilized to provide very high speeds at light loads, and automatic current step-back can give extra torque surges if required and this type of drive can be tailored to be very load-sensitive. A ship's up-roll can be internally sensed so that the drive will automatically ease up on the pull, or on the down-roll, raise the speed slightly. When well selected, the rotating power components give years and years of trouble-free service, and a sixteen-by-sixteen point master switch feels essentially "continuous" to the winch operator.

Today the Silicon Controlled Rectifier can do many of the same tricks, and do them with less physical volume and weight, and with truly variable speed control. The state of the SCR art has progressed to the point where users are reporting "failure rates" approaching "zero," once the initial shake-down period has been negotiated. Many of the electronic components actually improve and become more reliable with age, after the initial "burning in." The SCR type of DC winch drive should be considerably less costly than the Generator-Set type. By the nature of the components involved, it is easier to set up a complete electrical winch drive in the factory, and run the appropriate tests. When this is done, the on-board installation should become a matter of proper connections. This advantage, combined with the inherently clean environment for electrons inside a wire can greatly simplify the installation and start-up of a proper electrical drive, as compared to hydraulics.

A proper cost comparison between a properly installed winch with a closed loop hydrostatic transmission and the same winch with an SCR-DC drive system would be valuable and a life-cycle cost history over 20 years would be even more valuable. The initial cost of providing the electrical system is probably higher, but the difference must narrow when installation and start-up costs are factored in. For all the successful results provided by hydraulic machines, the costs of the additional energy, the transmission fluid, and the maintenance requirements may work to the disadvantage of the hydraulics. Given well done systems of both types, we would expect greater on-line reliability and less maintenance problems from the electrical system, although, admittedly this opinion is based on the older non-electronic forms of D.C. drives.

4.0 WINCH CONTROLS

The topside controls for a typical closed-loop hydraulic winch circuit involve a pump stroke control to vary the winch speed, a system to shift the reversing valve from HOIST to STOP to PAYOUT, and probably either an electric pushbutton or a PTO clutch control to get the pump running in the first place.

If the operating console is remote from the winch, there will also be valves for the drum brake and for the drum clutch. Most often the gear range selector is not remoted, since the pull/speed range for a given cast is not often changed.

From a control simplicity viewpoint, it would be elegant to have the AC/hydraulic power pack adjacent to the winch where the operator could move the pump stroker and the reversing valve themselves without remote hardware. Usually, however, remotes are necessary and levers, rods and cables seem less in favor then fluid types. The three types which are most common include air controls and two types of oil control. The simplest oil controls are yacht type manual "telemotors" displacing fluid along plastic tubes between sender handles and receiver levers. The pressurized-oil remote system is suitable for longer distances and requires a separate pump since most charge pumps would have difficulty trying to supply both the control volumes and the pressure needed to keep the main pump on stroke. This type of circuit is more complex than the usual WABCo. air valves and positioners, but if the vessel is operating in cold regions, it may be the only choice to avoid control freezing.

The available pressure-regulating oil control valves are small in size -- 1/8" diameter for Munson-Tyson, and 1/4" diameter for Greason, and consequently will NOT allow a high control oil flow. Positioners, strokers and cylinders require large volumes of oil in order to operate properly and there is a tendency for these units to respond slowly to the amounts of oil which a 1/4" valve will supply to and exhaust from the positioner. There may be a need in some installations to use the small control valves in a "Pilot" type circuit to apply bigger volumes to the positioners themselves. However, an application of "valves on valves" carries with it all the attendant leakage opportunities and increased maintenance problems of a complex system. Our preferenced would be the use of air positioners and air control valves. Action can be "right now" or regulated, as preferred, and the usual brands have a lot of service familiarity and parts availability.

5.0 GROOVED SHELLS AND FAIRLEAD DRIVES

Since there is an entire chapter devoted to grooved drum shells, we will note only a few comments regarding the requiremts for applying the nearly universal double-offset shell to a research winch. Our experience has been in the use of the Lebus shell in conjunction with the automatic traveling-head level-wind (or fairlead). When the head travel rate properly matches the groove pitches of the shell, the wire responds so beautifully that it is easy to wonder how the spooling could ever go wrong. The process, however, requires control of the spooling rate to the fourth or fifth decimal place, and also requires adequate tension on the line coming aboard.

Wire of 5/16" dia. is not the same as wire of 0.322" diameter. The two sizes result in different numbers of wraps and layers on a drum, in different pitch dimensions on the Lebus shell, and in different reduction ratios between the drum and fairlead diamond screw. In addition, a different filler plate should be provided for the metering sheave, and ideally, the throat contour of the suite of sheaves should differ. Therefore, one of the early tasks is to agree with a winch owner as to exactly what wire will be used, and then to obtain a sample. This sample should be sent along to the shell supplier, with your proposed fairlead ratio drive calculations, for his verification since simply measuring the accurate diameter of these wires is not easily accomplished in the usual shop. One means of reducing the effect of the flats and humps occurring in the wire is to lay up ten passes of wire next to each other and to measure all ten. In this way one's error should at least be reduced by that amount.

Before setting the fairlead drive ratio, it is necessary to decide on how many wraps of wire each layer will see. This can be calculated using the followig formula:

$$\frac{\text{Drum Face Width}}{\text{Wire Diameter}} = \text{Theoretical Wraps}$$

$$\text{e.g.} \quad \frac{30.000\text{" Face}}{0.250\text{" dia.}} = 120 \text{ wraps}$$

Wire does not spool against itself in a metal-to-metal contact situation and the grooves of the shell <u>must</u> allow a small air gap between the wraps. With a Lebus shell an air gap in the order of 1 1/2% of the wire diameter has worked satisfactorily and therefore, the 120 theoretical wraps are multiplied by 0.985 to reach a number of wraps equal to 118.2. Since this is an untidy result, it would be rounded to either 118 wraps or to 118 1/2 wraps in the final analysis. Rounding up to the 118 1/2 wraps leaves an air gap in the order of 1 1/4%, which should be more than adequate to ensure even spooling.

The trick, when applying a Lebus shell, is to remember that the number of DRUM TURNS per layer is ONE-HALF LESS than the number of wraps. Therefore, in considering the fairlead drive ratio, the HALF TURN is subtracted from the 118 1/2 wrap figue, to arrive at 118 TURNS OF THE DRUM per layer. Assuming a typical fairlead diamond screw with 7 1/2 diamonds at 4" pitch (to provide the 30" nominal head travel), the reduction ratio between the drum and the diamond screw is the ratio of the

$$\frac{118 \text{ DRUM TURNS}}{7\ 1/2 \text{ SCREW TURNS}} = 15.73333 \text{ to } 1.$$

However, the diamond screw may be driven, the numerical reduction ratio should be 15.73333 or as close to it as possible. It would be worthwhile to modify the 118 1/2 wrap assumption in the

search for the exact drive ratio since the closer the head travel matches the screw pitch, the less often the winch operator will have to stop, shift his fairlead head, and then continue. 118 1/2 wraps times the typical 30 layers on a winch drum gives 3555 individual "points" in a full drum, and the smallest displacement of each point accumulates to a visible mis-lay.

As an aside, it is this requirement for exact numerical ratios which suggests that variable fairlead drives ought not to be used. Multiple gear sets, as in a lathe or fine-toothed interchangeable sprockets, are a good means of changing the timing, of the diamond screw since these shift from one known ratio to another. Our experience with the variable speed devices is that they are almost automatically wrong, and that every well-meaning crew person and scientist knows that he or she can tweak it to just that illusive point of perfection.

When Lebus considers the correct shell pitch for a given wire, we understand that they put a tension on the wire when measuring it. This fact adds to the well known impossibility of spooling a loose wire and enforces the need for adequate tension at the shore side reel when bringing the wire onto the winch. Treating this small and delicate wire like a tug's tow-line by dragging it under a plank on which a fork-lift is parked does not really solve the problem. Ideally a proper reel truck would provide a retarder with tension measuring instrumentation so that the wire maker's book could be followed closely. In many practical situations things are not done so scientifically. At a recent "spooling-on" which we observed in a Seattle yard, there was only a reel cradle to accommodate the wooden mill spool. A pair of long 2 x 12 planks were found and wedged under the flanges, and two fitters were assigned to sit on the plank ends for the several hours involved. This application of leverage and friction worked well, and produced the lay illustrated in Photographs 1 and 2. In this particular case wire tension value of between 500 lb. and 1000 lb. was targeted for this 7/16" cable.

Many research drums spool the wire without benefit of Lebus shells with functional results. The care put into reel tension and the wrap-by-wrap hand control of the initial layer becomes even more important in this case and the fairlead spooling ratio would be seleted with a 5% to 6% air gap and more frequent manual head position adjustments would be expected. These conditions point out the need for a convenient hand clutch on the fairlead drive.

6.0 INSTRUMENTATION

The winch builder is concerned with providing the operator with information on the wire scope payed out, on the wire

speed, and upon the wire tension. This information is needed to assist the operator in putting his payload where it belongs so that the primary oceanographic data can be obtained. These three wire parameters are real time tools and the signals can be created most easily as the wire passes over a sheave. If the A-Frame or davit sheave is instrumented, the winch fairlead head can be a simple roller device only responsible for guiding the wire. A-Frame sheaves move around due to ship motion and the wire angle constantly varies, resulting in a high variance of readings. If the instrumented sheave and two guide sheaves are carried directly on the traveling fairlead head, the wire path is fully controlled and the compression load cell sees a constant vector. We have normally layed out the wire path with a 120° included angle under the metering sheave, since this gives enough wrap to minimize slippage, and since the load cell thinking is simplified by having the compression vector equal to the line tension.

Sheave size is important to the life of the wire -- seldom can the recommendations of the wire makers be accommodated within space and dollar constraints, but there is no denying the truth of Bigger Is Better where sheave selection is concerned. Our metering sheaves are of sandwich construction using replaceable "T-1" steel center plates to keep the pitch line of each wire at exactly the desired diameter. The thickness of these filler plates varies for each wire to keep the side cheeks lightly bearing on the wire for low-slip drive.

Markey applies four sizes of metering sheave, as follows:

Circumference	Pitch Diameter	Range of Wire Diameters
Half Meter	6.264"	1/8" to 3/16"
3/4 Meter	9.396"	3/16" to 1/4"
One Meter	12.529"	1/4" to 3/8"
1 1/2 Meter	18.793"	3/8" to 5/8"

These proportions range from 50 diameters to 30 diameters. The guide sheaves on either side of the metering sheave are somewhat smaller, and the pair of adjustable front rollers are between 3" and 5" in diameter. These rollers are cam-mounted so that the roller gap can be sized for the wire and so that the location of the gap can be correctly aligned with the front guide sheave. The angle of the wrap around these front rollers is usually very small.

Given the correctly sized sheave set, it is next necessary to select the transducers and display elements for Scope, Speed, and Tension. A compression load cell, such as a "Martin-Decker" is located between the sheave carrier and a fixed portion of the head. A local head-mounted tension dial is usually of the hyraulic type with a short connecting hose from the cell. For remote tension display a pressure-potentiometer

can be screwed directly into the cell, and tension read out on a properly faced voltmeter or strip chart recorder. For Speed and Scope, our preference remains with mechanical systems for local display, and with elecro-mechanical systems for remote display. By mechanical we refer to large-digit mechanical counters for scope, with a drive from the metering sheave via small stainless roller-type chain. And by mechanical we refer to mechanical tachometers such as the "Jones-Motrola," to display the line speed from the same roller chain drive. Acknowledging that wonders can be performed with electronic circuit boards and LED lamps, there is great "fixability" inherent in the mechanical hardware, and repair can often be affected with a can of spray lubricant.

The same roller chain will drive an Autosyn sender and a tach-generator to drive the remote counter and the remote voltage-type speed dial.

Display housings must be properly waterproofed -- this means machined box faces, heavy gasketing, and lots of screws. It also means heavy-duty, truly waterproof lamp power switches, and properly sealed counter re-set shafts if the display devices are expected to survive for any period of time.

Slip-ring assemblies have improved since the self-contained package units have replaced the separate finger type. Presently all that is required is a hollow main shaft, access holes in the drum flange wall and into the shaft center, suitably located wire clamps, and a saddle clamp to support the ring assembly. With the main shaft an integral portion of each winch drum, this sequence is not a problem.

7.0 CONSTRUCTION DETAILS

The winch drum is the heart of the Single Drum Research Winch, and there has been a constant increase in drum scantlings over the years. Small cold wire packed closely into a sun-heated drum, result in end forces which are tremendous. Flanges have become cone shaped, have been pushed off their securing welds at the barrel and the drum barrels have been stretched as a result of thermal expansion. Some years ago Max Silverman chaired a Wire & Drum seminar in Washington, DC, which lead to major increases in flange and barrel heft. Use of 3" plate flanges may sound absurd when looking at a sample of 1/4" wire, but if the spooling is to remain accurate and smooth, the machined inside surfaces of the flanges must remain parallel and at the design spacing. Presently we are hearing of a heavy-duty winch drum which was made heavier yet by rebuilding with 1 1/2" steel for the barrel tube.

Drum proportion is important, and here the smallest is not necessarily the best. A large core or barrel diameter reduces wire curvature, and shallows up the working depth of wire.

As the number of layers is reduced, the working radius variation is minimzied, and a few inches greater face width is well spent.

Drum interchangeability has been an extremely valuable feature allowing the substitution of pre-spooled spare drums when a change in program arises. This change-out requires a decent lifting facility and is best done on a calm day. The removal of the bearing cap, a split spacer collar and four bronze drive bolts allows the drum to be shifted off its pilot register and lifted out -- without disturbing the brake and clutch systems, the fairlead drive or the fairlead itself. The slip-ring unit stays with each drum, since the main shaft is integral. If the wire diameter has been changed, the suitable metering sheave filler should be inserted, and the proper sprocket installed into the fairlead drive train.

Base turntables have been provided for those installations where one winch must lead overboard at more than one location. With simplicity in mind these turntables have amounted to full diameter steel rings bolted to the deck foundation and providing for a loose guide register with the winch-base, and for the four heavy clamp blocks. Multiple grease fittings permit the introduction of a grease film, after which the winch is simply manhandled around to the new orientation. The auxiliary warping head has often been used to speed up this process.

The dual-range gearing on the single drum winch is totally enclosed with decades of life in mind, providing the gear and bearing lubrication receives careful attention. In addition to the usual oil-bath, a lube pump is sometimes used to move oil up to a gravity tray which feeds the copper lines leading to each bearing and each gear mesh. There remain ample opportunities for the deck enginer to apply his grease gun skills, and his steadfastness in this exercise makes all the difference in how the machine performs over the years. Grease is incredibly cheap compared to the labor and parts costs of a re-work.

Unless your vessel is a hydrofoil, the overall winch scantlings should lean toward the meaty side, rather than the skinny side. While avoiding design overkill and "wretched excess," the heavier winch maintains alignments and efficiencies, and the guards will still be there twenty or more years down the road. Sandblasting and inorganic zinc coatings are nearly universal in the scientific winch area, and well-applied zinc works wonders in keeping surfaces unpitted. The use of the inorganic zincs does involve real manufacturing costs, since the parts must be fitted and assembled, then removed, masked and coated. This double assembly appears to be well worth the initial investment. Often stainless pins and fittings are specified, with similar thinking in mind. Nuts, bolts and pins of stainless or monel or bronze will collect salt and detritous, and eventually freeze as solid as mild steel. It should be remembered that a torch will not melt out a stainless pin to

free a frozen linkage, and the next steps can be awkward.

Provision of a warping head costs little, and tnere are many on-deck situations where an extra live head is "as handy as a pocket in a shirt." Single Drum Research Winches are a proven and successful method of hoisting and storing oceanographic wires and conductor cables. As with all machinery, there are good design approaches, mediocre approaches, and downright silly approaches, and often there is a dollar implication involved in doing a project right. Having once done a project properly, a machine is created which remains a satisfaction to the operation and the builder for many years, and perhaps through several rebuilds and conversions as the needs change. Communications and feedback in the design, manufacture and operation of this machine are never good enough, and this represents an ever-available opportunity to do better. We wish to thank the sponsors and participants of this conference for the opportunity to air our thinking about the machine that amounts to the main battery on so many research vessels.

8.0 ORDERING INFORMATION

We have all seen ordering questionnairs which, if completed, will supposedly lead to the automatic creation of the optimum machine. Where a custom machine is involved, we would prefer to rely on lots of give-and-take with those involved -- inquiry, proposal, comment, revised proposal, order, detailing, approval submittal, comment, adjustment, and finally manufacture. Rather than formalized steps, an open process works the best and is much more satisfactory.

There are a few basic starting points which are nearly universal, and which can be listed as follows:

1) Wire diameter, or range of applicable wire diameters.

2) Scope of the wire, or wires.

3) Required speed of line recovery.

4) Projected payload weights.

5) Type of winch drive preferred.

6) The Amount of power assignable to the winch drive. Note that this often conflicts with item 3.

7) Location of the winch and its drive system on the vessel, particularly as regards deck space, line lead direction, access for control and maintenance, and the location of controls.

8) Requirements for Scope, Speed and Tension displays(s).

With this information in hand, the design and manufacture process can get under way.

CHAPTER 9

Double Drum Traction Winch Design

WAYNE JOHNSON
JOHN Le DET

1.0	THEORY OF OPERATION	9-2
2.0	WINCH POWER VS. TORQUE	9-6
	2.1 A.C. ELECTRIC DRIVE SYSTEM	9-7
	2.2 D.C. ELECTRIC DRIVE SYSTEM	9-8
	2.3 ELECTRO-HYDRAULIC DRIVE SYSTEM	9-8
	2.4 DIESEL-HYDRAULIC DRIVE SYSTEM	9-11
3.0	WINCH SIZING TO USER NEED	9-11
4.0	DRUM AND GROOVE DESIGN	9-13
5.0	MECHANICAL EFFICIENCY OF DOUBLE DRUM	9-19
REFERENCE		9-27

Double Drum Traction Winch Design

1.0 THEORY OF OPERATION

Although single drum winches have been in use since 2,000 B.C., the Double Drum Traction Winch has come into use more recently, or about the year 1900. Since their introduction many more single drum winches have been built than double drum traction winches. The single drum traction winch, or capstan, is also more widely used than the double drum traction winch. Most elevators use double drum traction winches, but these are normally hidden from public view. For these reasons, not many people are familiar with double drum traction winches as compared to single drum winches.

In addition to the use of conventional drums for pulling wire ropes, there are several types of friction drives for this purpose. In these units the movement of the rope is controlled by traction developed between the driving sheave or drum and the rope. With installations of this type, proper design should include a sufficient degree of holding or traction at the driving machine, generally through a sufficient arc of contact with the rope and an adequate coefficient of friction to prevent slippage while handling unbalanced loads.

The most common form of traction device known is the capstan, which was used extensively on sailing ships, and is still in use in many forms today (see Figure 9-1). Other forms and variations include warping winches, gypsies, catheads, etc., used with either vertical or horizontal driving shafts. The formula

$$\frac{T_1}{T_2} = e^{f\alpha}$$

governs the operation of any of these devices if acceleration of masses are ignored. In the above formula -

T_1 = Tension in tight rope (or loaded rope end)

T_2 = Tension in slack rope

e = Naperian base = 2.71828

α = Total angle of wrap in radians

f = Coefficient of friction between rope and drum

If we rewrite the formula to read $T_1 = T_2^{f\alpha}$, we note that the total developed output line pull T_1 is a function of the slack line side tension T_2, the coefficient of friction f, and the angle of wrap . Particularly note that if the slack line tension is brought down to zero, the output line pull T_1 also goes to zero. Because of this, some finite tension at T_2 must always be present to produce any output line pull.

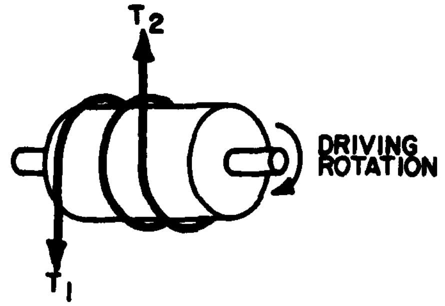


FIGURE 9-1 GYPSY OR SINGLE DRUM TRACTION WINCH

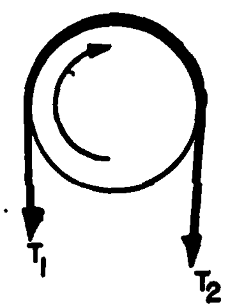


FIGURE 9-2 SINGLE DRUM TRACTION WINCH W/180° WRAP

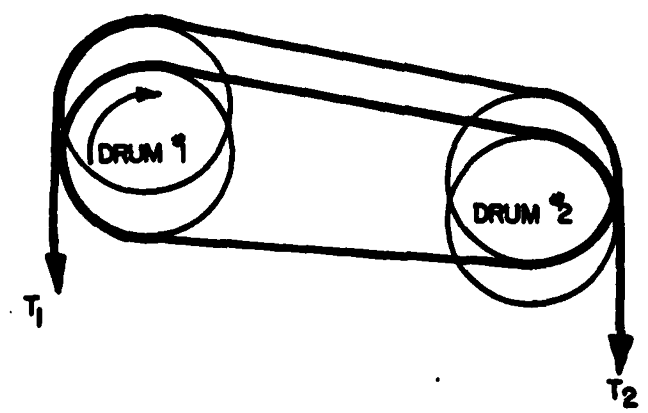


FIGURE 9-3 DOUBLE DRUM TRACTION WINCH

Figure 9-2 shows a special case of a single drum traction winch which utilizes only 180^{o} of rope wrap. This device requires a high rope-to-drum friction coefficient, and is commonly used in deep mine hoist work where a skip or cage is attached at each end of the rope to provide identical initial T_1 and T_2 loads. When ore is added to one of the skips for hoisting, the T_1 load of the skip at the bottom of the shaft increases. Note that since the empty skip being lowered counter-balances the weight of the skip being raised, the only energy that has to be put into the system is that necessary to lift the ore and minor frictional losses. After the skip being raised is dumped and the one then at the bottom is filled, the T_1 and T_2 forces reverse themselves, and the system is ready to lift ore from the second skip.

The double drum traction winch as shown in Figure 9-3 provides two (2) powered drums with multiple rope grooves with the grooves of the second drum offset one-half of a rope pitch in relation to the first drum. This allows a very high total rope wrap angle by reeving from drum to drum without producing the axial frictional movement of the rope which is inherent in the single drum traction winch or capstan. As shown in Figure 3 the total contact angle for the system is the summation of all contact on both drums since both drums are powered. This system works particularly well when low rope to drum friction coefficients are encountered.

A special case of the double drum traction winch has power on only one of the two drums with the second one being used as an idler. Since about half of the total rope to drum contact angle is lost with this system, the design is relatively ineffective and is reserved for use in only very special applications.

The most obvious difference between the conventional winch and traction winch is that the traction winch requires some means of pulling the cable off of the drums (see Figure 9-4). For wire rope, a storage drum usually serves the dual function of providing a back tension as well as cable storage. When soft rope is used, a tension wheel or the weight of the rope can provide back tension on the winch and the rope can be stored in a storage bin similar to the type used for anchor chain in lieu of a storage drum. This works especially well with braided rope. However, the purpose of this study is to examine double drum traction winches for use with wire rope. The emphasis will be placed on the double drum wire rope traction winch, since the storage winch is essentially a conventional winch modified to produce the constant tension required to provide the T_2 loading at the traction winch.

Friction drives, rather than drums, are used for reasons of economy, safety, or sometimes necessity. Where a rope is used on an endless conveyor system, or rope ski tow, the spliced rope cannot be driven by any other method. On modern

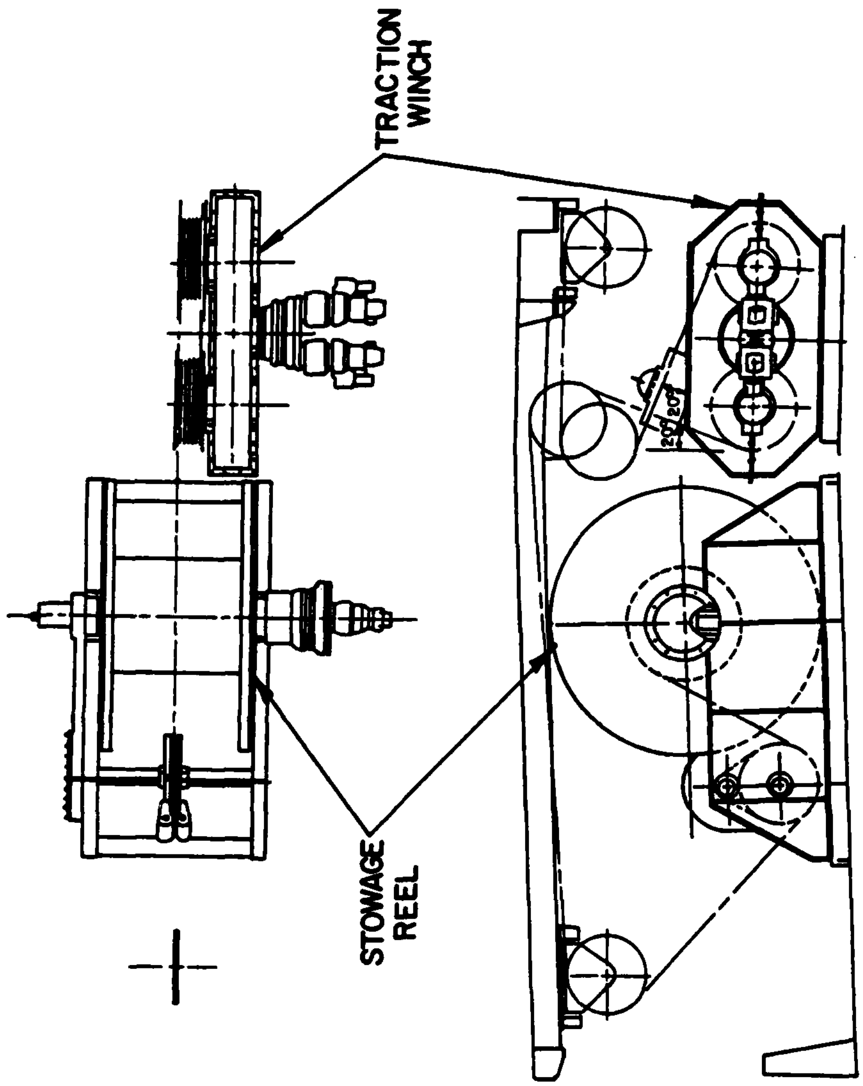


FIG. 9—4 DOUBLE DRUM TRACTION WINCH

passenger elevators the traction drive is an important safety feature, since it permits slippage to occur between ropes and drive sheave in the event of failure of the safety devices designed to prevent overtravel at the terminal landings. On some installations, a friction type drive is a matter of economy or necessity, since a drive sheave is smaller, lighter and requires less space than a winding drum capable of sorting rope for an appreciable length of travel. Also, since the traction drum pitch diameter is constant, the change of load and speed capacity caused by multi-layers of rope on a conventional winch can be eliminated. This is important when long line lengths are encountered such as in oceanographic work.

2.0 WINCH POWER vs TORQUE

The useful output horsepower produced by any winch follows the conventional formula

$$HP = \frac{PV}{33000}$$

where P is the total load being lifted in pounds and V is the lifting velocity in feet per minute.

This same output horsepower can be expressed in terms relating to the drum as

$$HP = \frac{TN}{5250}$$

where T is the Torque at the drum in foot pounds and N is the drum speed in revolutions per minutes.

With P and V known and the drum pitch diameter selected (see Section 4-0) Drum Torque can be determined by the simple formula

$$T = P \times \frac{D}{2}$$

where D is the pitch diameter in feet. Drum speed can also be determined as

$$N = \frac{V}{\pi D}$$

where D is the pitch diameter in feet and π is the constant 3.14159.

Since most power sources usually produce power at relatively high speeds, the torque produced is usually too low to apply directly to the traction drum. Because of this, some form of torque multiplication must be applied, and this usually is in the form of gear reduction. (More information on gear reductions is given in Section 5-0)

Besides the basic drive train or gear reductions used on a traction winch, the power source of the winch can vary substantially. The four categories which can be easily adapted are as follows:

- o A.C. Electric
- o D.C. Electric
- o Electro-Hydraulic
- o Diesel-Hydraulic

In any case, several features are desirable, and are available in each category. The importance of one feature over the others is sometimes the governing factor when choosing a power system. One important characteristic of traction winches must be remembered when choosing a power system. That is, the line pull of the winch is not affected by the amount of cable payed out, as is true of a conventional single drum winch. When used with traction winches, the various power systems should have the following features:

a. Stepless variable speed control in both direction.

b. Variable motor torque which can be utilized at maximum potential at low speeds and at stall for extended periods of time.

c. Ability to hold a load stationary and lower it at a controlled rate.

d. Adjustable torque control to limit maximum line pull.

e. Capability of operating with "failsafe brake."

f. High light line speed to handle unloaded or partially loaded line when necessary.

2.1 A.C. Electric Drive System

For use with a double drum traction winch, there are two types of A.C. Electric Drive Systems which meet most of the criteria listed above. They are the variable or adjustable frequency drive and the stepless controlled wound motor drive.

1a. Current controled adjustable frequency drive systems can be used with a standard "Nema B" curve squirrel cage induction motor. By adjusting applied voltage and frequency, the speed is adjusted from zero to full RPM, without causing current

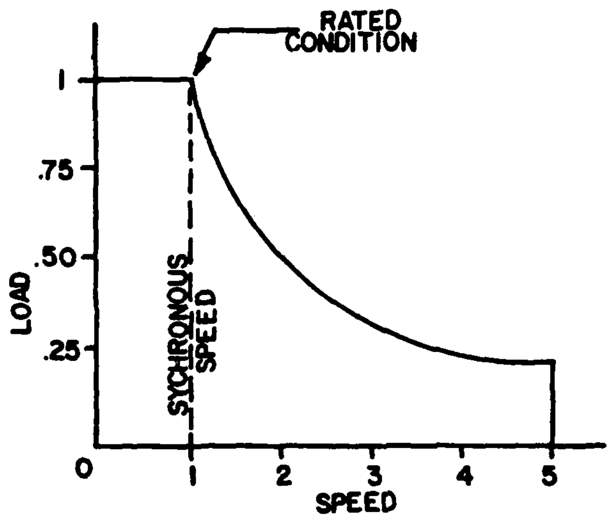
overload in the motor. The current is proportional only to the load induced into the motor. Therefore, the maximum allowable current can be present, and is adjustable to meet operating requirements. Light line speeds as high as five times rated can be obtained depending upon the design of the drive package (see Figure 9-5).

1b. Wound Rotor Motor Drives have been used for many years to operate conventional winches, and can be used to power traction winches. Stepless speed controls have been developed which eliminate the objectionable "steps" originally used to control these motors. Most of the desirable features listed above can be met. These motors can produce high torque at slow speeds, with stall capabilities for relatively extended periods, as well as slightly higher speeds at reduced loads (see Figure 9-6). Large resistor banks are used to absorb current to adjust starting torque, starting current, and running slip. Special "Nema X" design motors with high temperature insulation are used with this system, and have proven to be reliable through years of use. Marine applications such as towing winches on tugboats and anchor winches on offshore oil field drilling rigs have indicated their dependability. Lowering loads at low speeds sometime becomes a problem with a wound rotor drive. Note, also that full rated horsepower is consumed all the time that rated load is applied to the T_1 line is not used in doing work is consumed in the resistor banks.

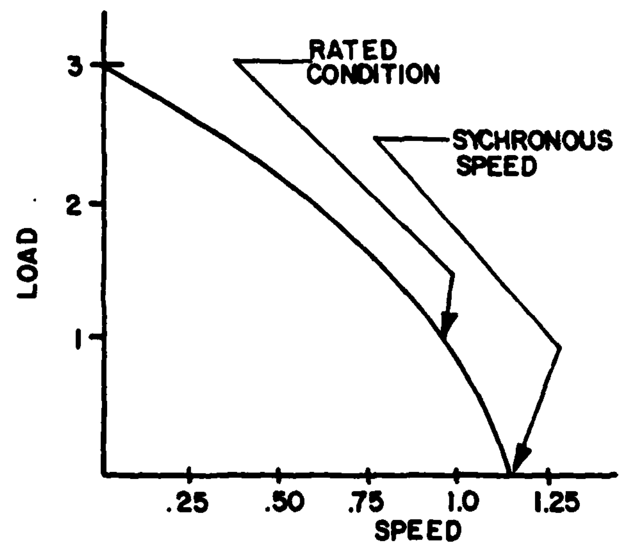
2.2 D.C. Electric Motor Drive Systems have also been developed in recent years which are much less expensive and cumbersome than the old Ward-Leonard System. Bu using SCR (Silicon Controlled Rectifier) motor controllers, A.C. current can be converted and controlled for D.C. Motor operation. Using a shunt wound D.C. Marine Motor, with high temperature Class F insulation, infinite manual speed control can be achieved by controlling armature voltage below base speed (constant torque), and adjusting motor field strength above base speed (constant horsepower) (see Figure 9-7). A light line speed of four times rated or base speed is a practical limit. Adjustment of the armature current will regulate motor torque (line pull) to any desirable limit. When the motor operates above base speed, the output torque will automatically decrease to provide constant horsepower characteristics. Many variations of this control can be obtained with options to meet any normal requirements. Note that D.C. motors can operate at stall conditions for very limited periods of time; usually less than one minute.

2.3 Electro-Hydraulic Drive Systems have been successfully used in marine environments for many years and continue to gain popularity over electro-mechanical drives (A.C. and D.C.). The ability to change speeds rapidly, free-wheel, change direction quickly, change torque settings easily, and withstand unfavorable enviroments in many instances exceeds the capability of the electric drive systems. Utilizing standard "on the shelf" com-

TYPICAL LOAD vs. SPEED CURVE WITH UNITY SPEED AND LOAD AT RATED CONDITION.

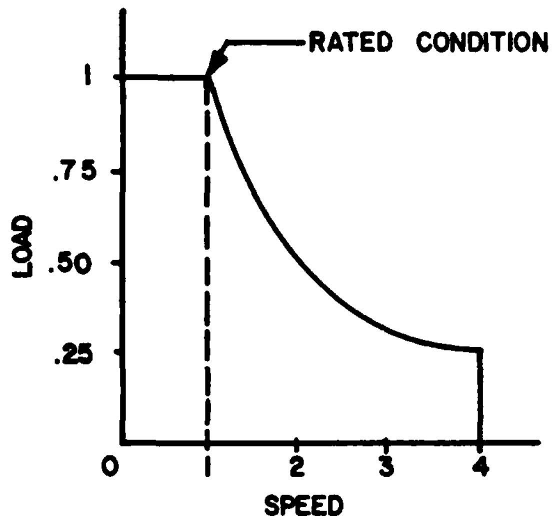


A.C. ADJUSTABLE FREQUENCY FIGURE 9-5

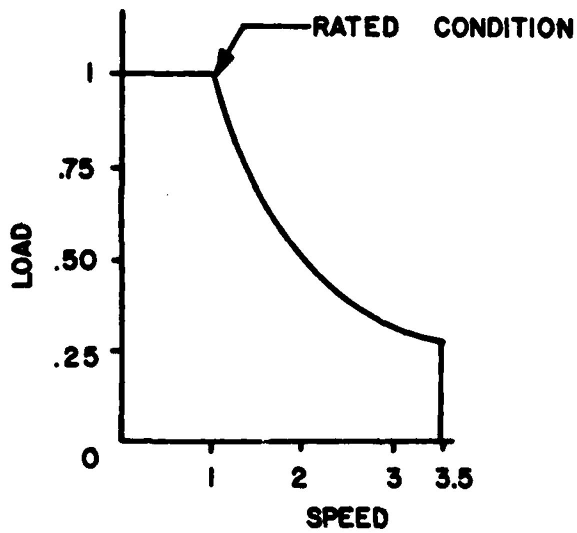


A.C. WOUND ROTOR FIGURE 9-6

TYPICAL LOAD vs. SPEED CURVE WITH UNITY SPEED AND LOAD AT RATED CONDITION.



D.C. ELECTRIC FIGURE 9-7



DIESEL HYDRAULIC OR ELECTRIC HYDRAULIC
FIGURE 9-8

ponents as opposed to special electrical equipment reduces initial cost and decreases maintenance and spare parts problems. Noise levels are often higher than electric drives, but newer equipment designs are becoming more and more quiet. A good feature is the ability to couple hydraulic motors directly to the winch drum to eliminate all gear if so desired, or reduce the number of gear reductions to a single reduction with a low or medium speed motor. Hydraulic pumps and motors have little rotating inertia which allows them to keep up with rapidly changing load conditions. A maximum light line speed of approximately 3.3 times rated speed is a practical limit for a fixed stroke hydraulic motor and variable stroke pump (see Figure 9-8). If the motor is also of the variable stroke type, the total light line ratio to ratio condition could be as high as the 8 to 10 range.

2.4 Diesel Hydraulic Drive Systems become attractive for powering traction winches when high horsepower levels are required on ships with relatively small electric generating systems. A completely portable winch system can be built and utilized on any vessel regardless of electrical service available on the vessel. Size and weight may increase slightly over electro-mechanical or electro-hydraulic systems, but initial cost is about the same. Actual operating costs are less with diesel units if generator systems on boats are not over designed to operate "portable" electric drive winches. Noise levels for diesels will be higher than electric or electro-hydraulic systems, but much is being done to reduce these noise levels. Almost all of the characteristics of electro-hydraulic drive systems apply to diesel hydraulic, with the exception of the inability to utilize regenerative braking of the engine to retard heavy loads being lowered dynamically. The addition of counterbalance valves, or a regenerative hydraulic braking system, can bring about control of even the heaviest loads.

The speed vs torque characteristics are identical to that shown for the Electro-Hydraulic System.

3.0 WINCH SIZING TO USER NEEDS

Whenever a winch is required to accomplish a task, several criteria must be examined to determine exactly what size winch is best suited for the job. The basic steps to take are as follows:

- Determine the maximum load to be lifted.
- Choose a cable size which will give the desired safety factor
- Determine maximum line pull required.
- Determine the rated line pull required.

- Determine the length of cable required for the water depth
- Choose a reasonable rated line speed for hauling. (Average Speed).
- Choose a maximum light line speed.
- Determine horsepower requirement at rated line pull and speed.

To determine the maximum line pull, add the weight of the load, plus the weight of the cable, times 1.25 for pullout, times .872 for bouyancy. Max. Line Pull = [(Cable Length) (Weight/Unit of length) + Load] X 1.25 X .872

Example:

Max. Line Pull = [(10,000 Meters)(1.57 Lbs/Meter + 5000 Lbs.] X 1.25 x .872 = 22,563 Lbs.

To determine the rated line pull, add the weight of half of the cable, plus the weight of the load, times .872

Example:

$$\text{Rated Line Pull} = \left[\left(\frac{10{,}000 \text{ Meters}}{2}\right) (1.57 \text{ Lbs./Meters}) + 5{,}000 \text{ Lbs.}\right] \times .872 = 11{,}205 \text{ Lbs.}$$

Choose an average speed and maximum speed.

Example:

Rated Speed = 500 FPM Maximum Speed = 600 FPM

To determine the horsepower required, use the following horsepower formula:

$$HP = \frac{\text{Rated Line Pull (pounds)} \times \text{Rated Line Speed (feet per min)}}{\text{33,000 pound-feet per min per horsepower} \times \text{drive efficiency}}$$

Example:

$$HP = \frac{(11{,}205 \text{ Lbs.})(500 \text{ FPM})}{(33{,}000)\ (.7)} = 243 \text{ Horsepower}$$

Note that the 70% efficiency used is an approximate overall system efficiency, and this will vary somewhat with the drive system actually used.

In order to utilize existing or available equipment, line pull, line speed, and cable size can be changed to meet the horsepower available.

4.0 DRUM AND GROOVE DESIGN

Since in oceanographic work we are faced with the problem of handling very long wire ropes, the drums will be subject to a higher proportion of wear contacts than the average installation. Because of this, the drums should have a fairly hard surface which, unfortunately, has a tendancy to lower the friction coefficient f. To counteract this, we need as much contact angle as possible, so we will confine our discussion to multiple reeved double drum fraction winches with both drums being powered.

The formula generally used for designing a friction drive, where the rope operates U grooves (see Figure 9-9) or on a spool is

$$\frac{T_1}{T_2} = e^{fn\pi}$$

as shown at the top of Table A. This is derived from the general formula shown near the bottom of the page. (Note that this formula differs slightly from that shown in Section A in that n is substituted for friction angle α. The term n is the number of active 180° rope contacts and the term π converts these one-half wraps to the total angle in radians. This formula assumes that all contacts are of 180° magnitude which is not always the case in regard to the two end contacts. If the end contacts are not 180° the angle must be adjusted accordingly. The general formula takes into account a V groove, and also the effect of centrifugal force, but this latter factor is seldom of significance on wire rope installations (see Figure 9-10).

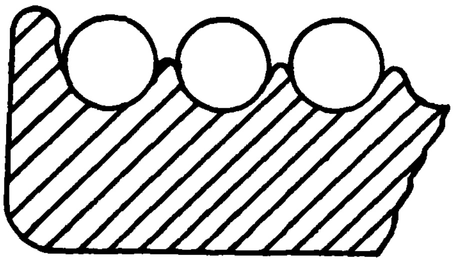
Two important points should be stressed regarding friction drive design. One is that in addition to the coefficient of friction, traction depends on the arc of contact and not the size of the driving sheave or drum. Therefore, if the arc of contact is two half wraps, or 360°, traction would be the same with a 50" diameter sheave as with a 100" diameter sheave. It makes no difference that the length of rope in contact is twice as long with the larger sheave.

The other point is in regard to the tensions T_1 and T_2. These should be calculated on the two sides of the system, at the drive sheave, for the condition where the ratio

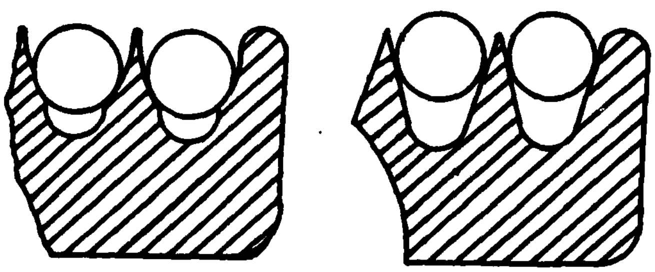
$$\frac{T_1}{T_2}$$

is maximum. Calculating this ratio for static loads is not correct because, during acceleration and deceleration, additional forces are invovled. For instance, when the load on the heavier side is moved and accelerated toward the driving machine, the rope stress on this side will be increased by the acceleration stress,

$$F = \frac{W}{G}\ a.$$



DOUBLE WRAP TRACTION DRIVE
FIGURE 9-9



SINGLE WRAP TRACTION DRIVE
FIGURE 9-10

At the same time, the stress on the off-going, lighter side will be decreased by another stress, and this can result in a decidedly greater

$$\frac{T_1}{T_2}$$

ratio.

Friction drives, unlike drum winding installations, are not positive drives. With drum winding, the load at the end of the rope always travels in direct relation to the drum movement, making proper allowance for rope stretch. An indicator driven by the drum can be depended upon to show the position of the load.

This is not the case with a friction drive because of creepage. Creepage is the slight movement which occurs between rope and drive, resulting from the elastic properties of the rope. Because of the difference in tension on the two sides of the system, the rope must change in length while in contact with the driving sheave, and this causes the relative movement. For each complete cycle of operation, except when $T_1 = T_2$, there will be a difference between rope and sheave travel. If rope and sheave are marked at the start, the marks will be in a different relative position at the end of the cycle.

Since with friction drives, the rope and the loads attached to it are driven by the traction developed between the driving sheave and only the outside wires of the rope strands (and, sometimes only a few of these outside wires), the effect on the rope is quite different than on a drum winding instalation. Larger outside wires will generally withstand this type of fatiguing action better then smaller wires and the most suitable rope construction should be chosen with this in mind. The pinching action of V or other shaped grooves is even more abusive and the severity of this action is often the determining factor in the life of a rope. Also note that the pinching action of the V groove increases rope contact pressure which increases the

$$\frac{T_1}{T_2}$$

relationship.

To operate unbalanced loads properly, the traction relation must be equal to, or greater than, the ratio of the tension in the tight rope to the tension in the slack rope

$$\frac{T_1}{T_2}$$

The traction relation is a function of the number of half wraps of rope contacting the driving sheave or sheaves, the coefficient of friction, and the shape of the gooves. For a rope operating on a flat surface or in a U-groove, the traction

TABLE A (Reference 1)

WIRE ROPE FRICTION DRIVES

$\frac{T_1}{T_:} = e^{fn\pi}$ (see note below) $T_1 - T_: - P$

Where T_1 = Tension in tight rope
T_2 = Tension in lack rope
e = Naperian base = 2.71828
n = No. half wraps on driving sheave or sheaves
f = Coefficient of friction
P = Tractive or Driving Force

COEFFICIENTS OF FRICTION (f)

	Greasy Rope	Wet Rope	Dry Rope
Grooved iron or steel sheave	.070	.085	.120
Wood lined sheave.	.140	.170	.235
Rubber and leather lined sheave.	.205	.400	.495

Values of $e^{fn\pi}$ for listed values of "f"

n Half wraps	.070	.085	.120	.140	.170	.205	.235	.400	.495
1	1.25	1.31	1.46	1.55	1.71	1.90	2.09	3.51	4.73
2	1.55	1.71	2.12	2.41	2.91	3.62	4.38	12.3	22.4
3	1.94	2.23	3.09	3.74	4.96	6.90	9.16	43.4	106.
4	2.41	2.91	4.52	5.81	8.47	13.1	19.1	152.	501.
5	3.00	3.80	6.58	9.02	14.4	25.0	40.0	535.	2390.

6	3.75	4.96	9.60	14.0	24.7	47.7	83.9	1875	.....
7	4.66	6.48	14.0	21.8	42.1	91.0	175.	.....	.....
8	5.81	8.47	20.4	33.8	72.0	175.	365.	.....	.....
9	7.24	11.0	29.6	52.5	122.	330.	765.	.....	.....
10	9.02	14.4	43.4	81.5	210.	625.	1600.	.....	.....
11	11.2	18.8	63.4	126.	360.	1200	.....	.....	.....
12	14.0	24.7	91.9	196.	610.	2280	.....	.....	.....
13	17.5	32.2	135.	305.	1040	.....	.....	.....	.....
14	21.8	42.1	196.	475.	1790.	.....	.....	.....	.....
15	27.2	54.6	285.	740.	.....	.....	.....	.....	.....

Note: The above formula derived from the general formula

$$\frac{T_1 - T}{T_2 - T} = e^{fa/\sin\frac{\omega}{2}}$$

where T_1, T_2, e, and f are as above and $t = \frac{WV^2}{g}$

(where t is the effect of centrifugal force)

W = weight of rope in lbs. per ft.
v = velocitry of rope in ft. per sec.
g = 32.2 ft per sec.^{2}
a = angle, in radians, subtended by arc of contact.
ω = included angle of groove sides when rope is supported by the sides, i.e. V-groove. When U-groove is used $\sin\frac{\omega}{2} = 1$ $\omega = 30^\circ$ to 45°

For speeds up to about 25 ft. per sec., t may be neglected.

relation is equal to

$$e^{fn\pi}$$

and values of this relation are given in Table A for different numbers of half wraps and different coefficients of friction.

The tractive or driving force is dependent on the difference between the tensions in the tight and slack ropes (T1-T2). Power to drive the system must be sufficient to supply this force at the required speed. Note that a finite value of T2 must be present to develop any tractive force. Without it, the rope would just slip on the drums.

The rope usually leads from the load to the first groove in the back drum or sheave, and from there to the first groove in the front sheave. From this it leads to the second groove in the back sheave, then to the second groove in the front sheave and so on, until the necessary number of wraps have been made. From the last groove in the back sheave, the rope goes to a counterbalancing load or counterweight. The rope is reeved between the drums of sheaves so that no reverse bends are encountered. In other words, the rope should not be reeved in a figure 8 configuration between the drums if satisfactory rope life is to be expected. Drums or sheaves should be spaced far enough apart so that the various parts of the rope in the the reeving will not scrub severely on the sides of the grooves as the rope leads from one sheave of drum to the other. It is preferable to keep the fleet angle of the rope less than 1^{o} 30' if this abusive action is to be kept to a minimum.

Since considerable bending is encountered on a winding machine of the type where the rope is reeved in a number of parts, it is important that the drums of sheaves be of liberal diameter. This equipment should be at least 40 times the rope diameter.

In order to avoid building up unequal tensions or differential action between the various wraps of rope on the machine, resulting from heavy or uneven groove wear, slip-rings can be used in the driving drum. With slip-rings, any tendency toward the occurance of slippage due to unequal tensions will result in a relative movement between the rings and the drum, rather than between the rope and the grooves; in this way considerable rope wear and fatiguing action can be avoided.

Two lines of thought exist when it comes to drum material. One is to protect the drums, the other is to protect the rope. The first approach emphasizes that the rope should not wear the drum grooves if normal service is expected. Therefore the material in the groves should be harder than the material in the rope, or the flexibility of the rope must be modified. For any given installation where the rope stress is established and the size of the rope determined, the diameter and

the material of the drum have a great bearing, both on the life of the rope and on the wear of the drum.

The second approach points out that the driving drum grooves on friction hoists can be lined with materials which provide a higher coefficient of friction than can be obtained with iron or steel. Originally wood liners were employed, then metal such as aluminum, then leather. Lately plastics have been used. Special plastic materials are being developed which have the necessary friction properties and good resistance to wear.

Since the groove liners have been softer or less wear-resisting material than steel, there has always been some concern regarding liner life when selecting the type of rope to use. The problem has been complicated by the fact that lubrication must be controlled carefully in the new rope, as well as in field application, to avoid the possibility of slippage. Sometimes rope constructions have been chosen primarily because of their ability to provide best economy, considering their original cost and the time they could safely be left in service.

With newer plastics and further development, it may be possible to improve these conditions by changing the hardness or wear properties of the liners to suit the need. In theory the coefficient of friction between two materials determines the available traction when considered together with the shape of the groove and the arc of contact. However, in practice, it is known that when a material deforms due to its elasticity, this also has some effect. Therefore, it might be possible to vary the liner material to suit the rope, designing for maximum rope service without detracting from the expected life of the liners. Rather than allow rope with a rougher exterior to embed deeper into the soft liners, a tougher liner material might be used to offer greater resistane to wear and at the same time allow sufficient embedment to develp necessary traction.

5.0 MECHANICAL EFFICIENCY OF DOUBLE DRUM SYSTEM

In a double drum traction winch the most economical and efficient final drive connecting the two drums consists of a spur gear connected to each drum with a common pinion driving both drum gears. Each drum can be supported by either two or three anti-friction bearings. Normal losses per loaded anti-friction bearing can be taken at .5% per bearing, but in the case of the double drum traction winch, the multiple reeving required tends to load up the drum shaft bearings to a considerably higher value relative to the T_1 pull developed. Because of this, the efficiency loss per bearing can be taken at 1% per bearing. The loss across the spur gear mesh can be conservatively taken at 2%, so the total efficiency for the final reduction would be the summation of efficiencies of the gear mesh and bearings involved. This would produce a final drive efficiency of 96% for the two bearing shaft [(1-.02-.01X2) X100%] or 95%

[(1-.02-.01X3) X 100%] for the three bearing shaft.

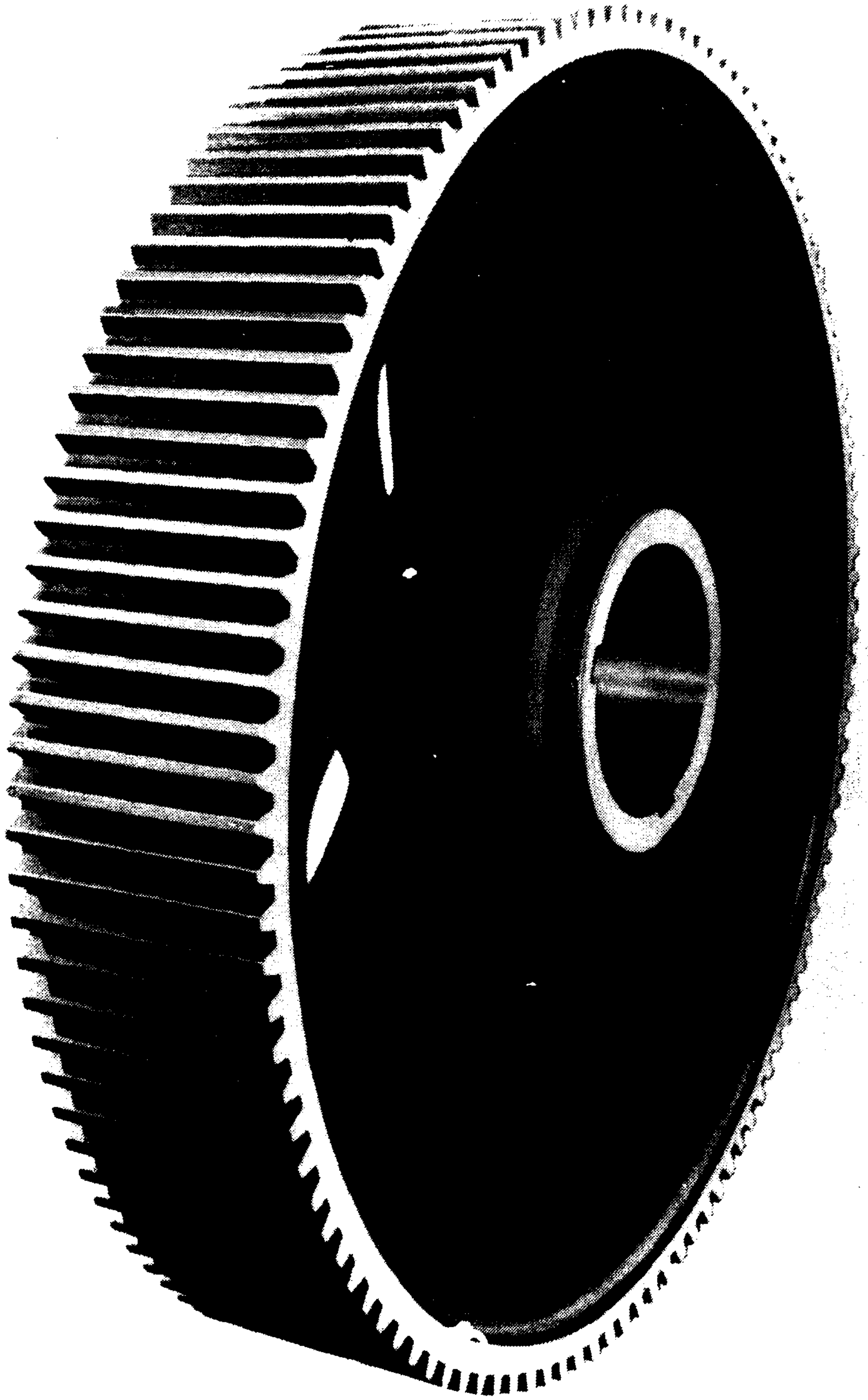
Average sleeve bushings produce about 2% loss each, so if sleeve bushings were used these efficiencies would change to 94% and 93% respectively. Note that the above efficiencies need only be applied once as only one-half of the power is transmitted to each drum, and therefore the total frictional loss will be accounted for if the efficiency value is applied one time to the total power transmitted. Sleeve bearings are sometimes used on the drum shafts to provide higher shock loading capabilities, but provide a penalty in efficiency as noted above.

Anti-friction bearings are usually preferred on intermediate and high speed shafts because the 0.5% efficiency loss can be used at these locations and heavy shock loading is not normally encountered away from the load end of the gear train. Helical or Herringbone gears could be substituted for the spur gears mentioned with the same gear mesh efficiencies. These gears run quieter than spur gears but at the penalty of added cost. Note also that Helical gears also create shaft end thrust forces that must be accounted for in the design of the winch.

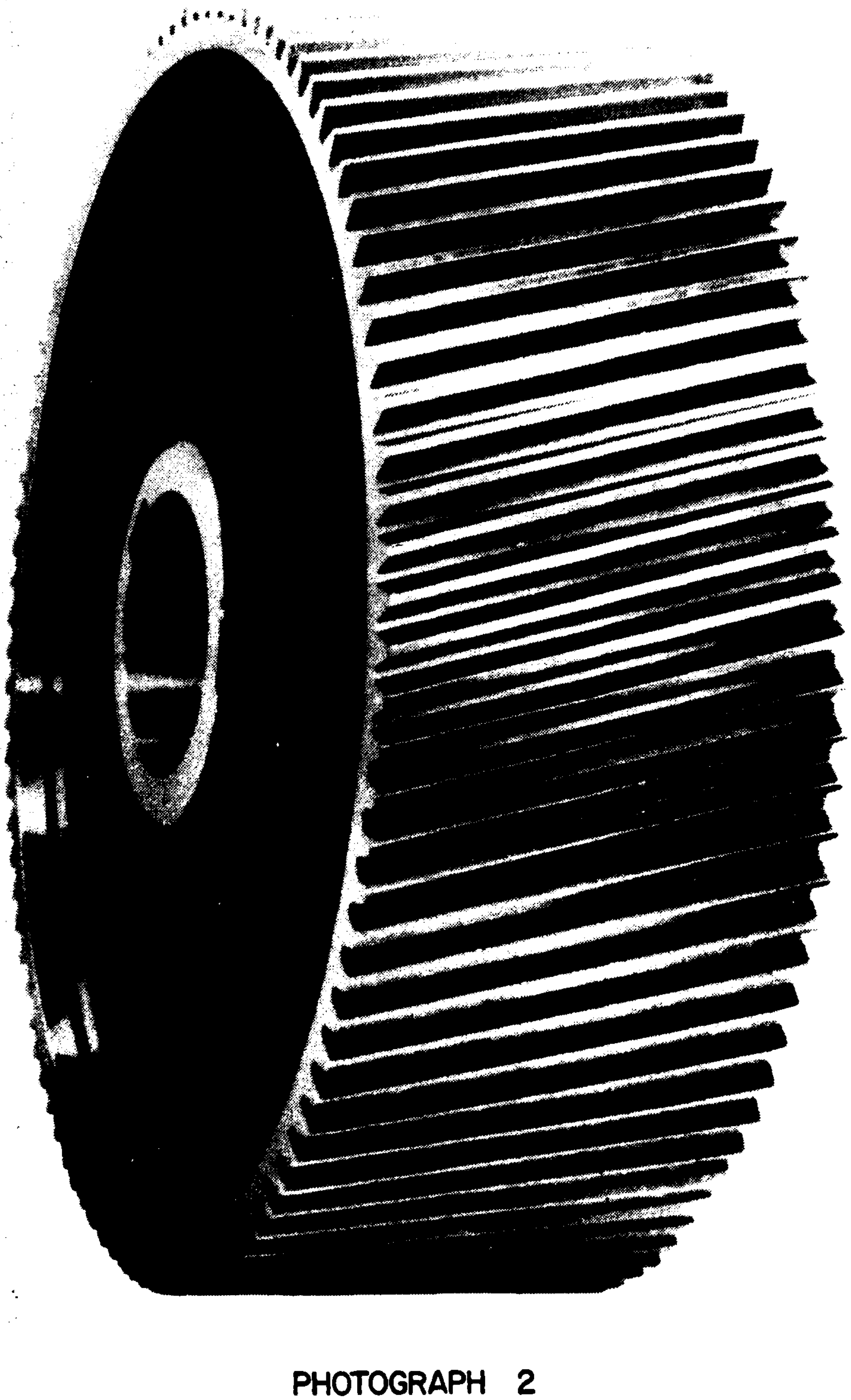
After the final drive or low speed gear set, the drive train can consist of a variety of choices. A low speed, high torque hydraulic motor can drive the pinion gear transmitting power to both drums, or a separate motor can be used to drive each drum. This would be the simplest drive train, with no additional gears required at all, but the hydraulic motors driving each drum separately could be connected together hydraulically to prevent slippage of the drum having the least resistance.

Another method of driving the low speed pinion gear is to use a spur, helical, or herringbone (or combination thereof) gearbox incorporating single, double or triple stages of gearing. Horsburgh and Scott published efficiencies of single, double and triple stage gearboxs as 98.5%, 97%, and 96% respectively. These values are for ideal continuous running conditions and should probably be lowered by one or two percent to account for such things as churning cold oil for intermittent applications. With this type of gear drive, any type of motor could be used from low speed (50-200 RPM), to medium speed (200-1500 RPM) as well as high speed (1500-6000 RPM), depending on the reduction that is used.

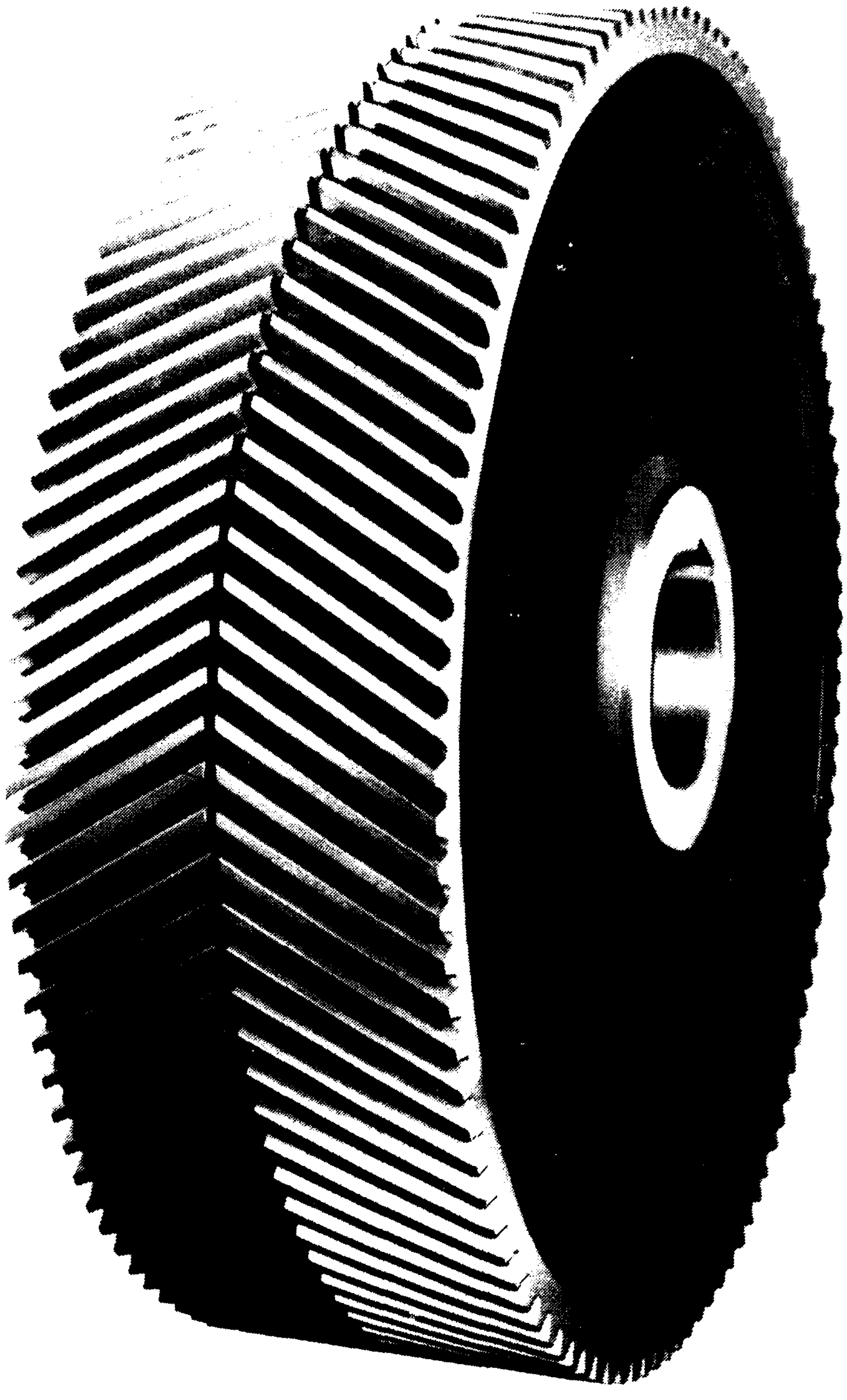
Besides spur, helical or heringbone gearboxes (Photographs 1-4), planetary gearboxs are also used with comparable reduction ratios but slightly lower efficiencies (about 1% Less) for equal reductions. Planetary gearboxes are usually more compact and adaptable to hydraulic drives than helica or herringbone boxes, and offer many "plug in" conveniences such as multiple inputs, SAE flange mounts, and the ability to be "stacked" or added to conveniently. Because of these characteristics, planetary gearboxes, developed for mobile equipment, can be much more economically adapted for winch drives than industrial type



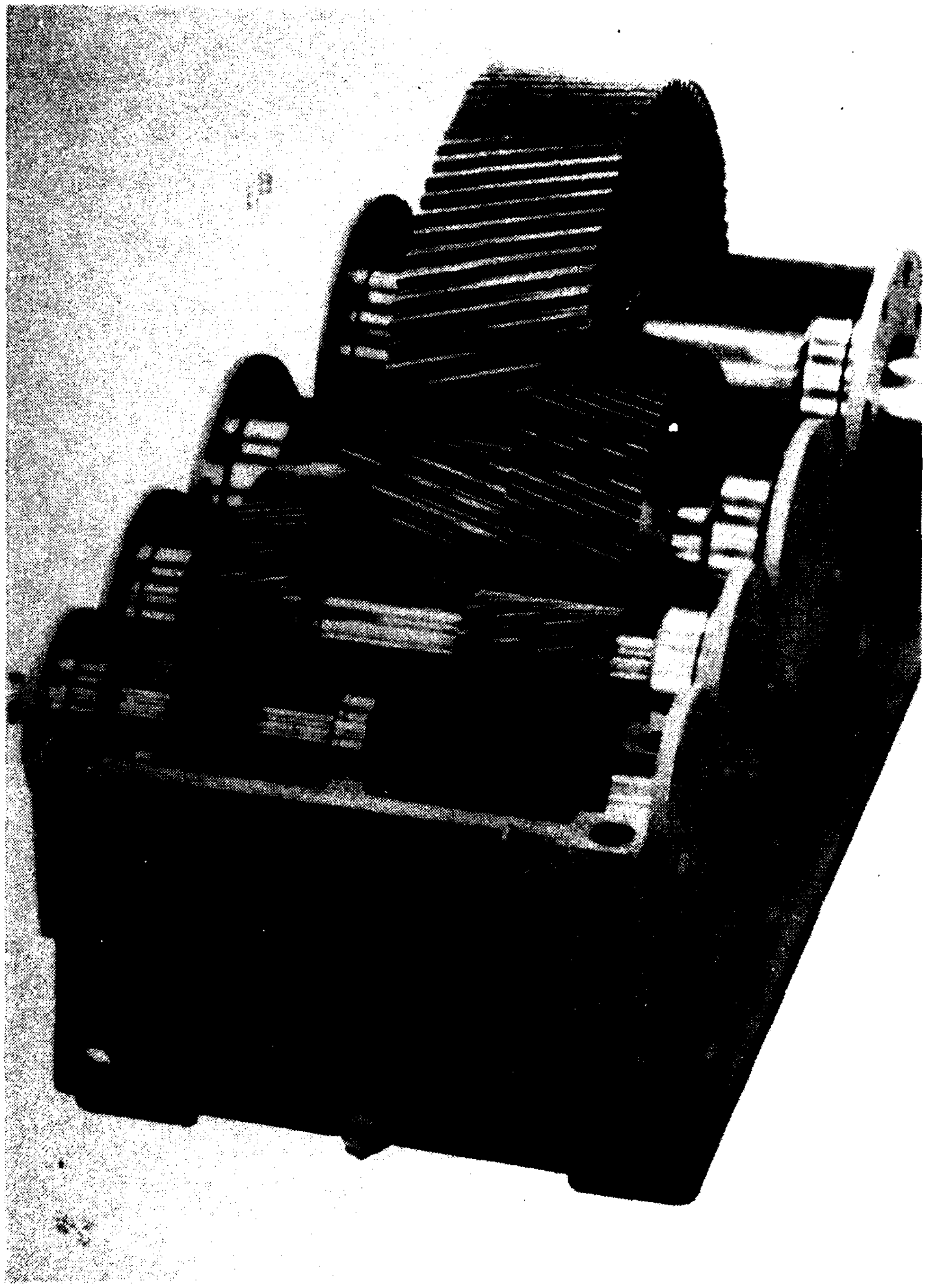
PHOTOGRAPH 1



PHOTOGRAPH 2



PHOTOGRAPH 3



PHOTOGRAPH 4

gearboxes or special design gearboxes.

Chain drives are sometimes used due to physical location of components, with reductions limited to approximately 6:1 per sprocket set, and efficiencies of about 96% per reduction.

Worm drives (Photograph 5) are relatively inefficient (from 35 to 90%), and are normally not used in traction winch drives.

Hydraulic torque converters are 80 to 85% efficient at best over a relatively limited speed range with efficiencies decreasig rather drastically with any appreciable amount of slip. Transmissions that may be connected to the torque converter will produce additional losses equivalent to the gearbox efficiencies mentioned previously.

Hydraulic torque converters can be used with electric motors or diesel engines to cushion the effect of high momentum in these motors, while being used to multiply torque. Output torque increases as the load increases, and output speed decreases as torque and load increases. Modulating torque converters with variable speed control are also available.

Electic motors and diesel engines provide net output horsepower and efficiencies are normally accounted for beyond that point.

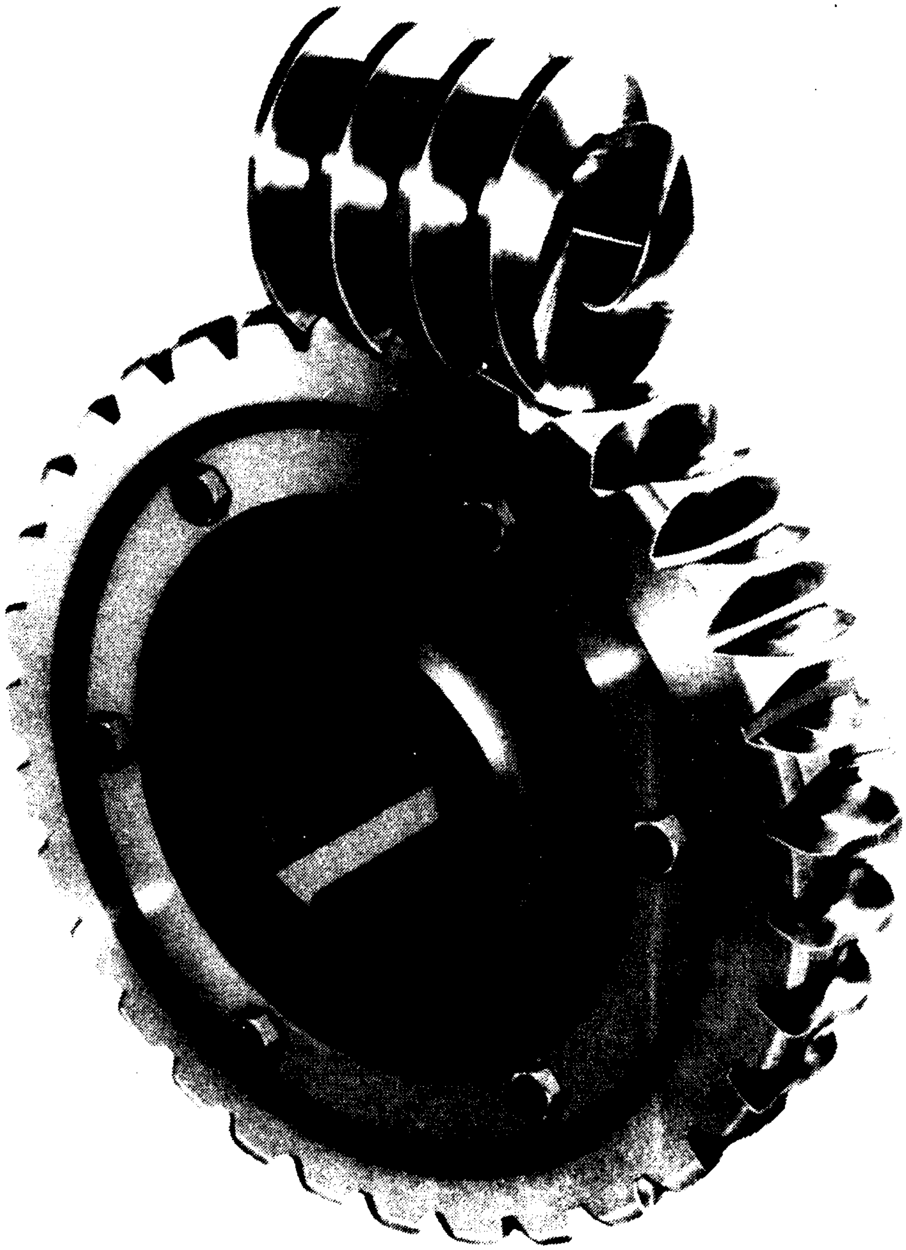
Axial piston hydraulic pump and motor efficiencies will vary from 85 to 90% each in their normal rated operating range, and they will drop slightly at both low and high speeds as well as at very low and very high pressures.

A typical overall efficiency of a double drum traction winch which is powered either by an electric motor or diesel engine through an axial piston pump and motor combination which in turn drives a two stage planetary gearbox whose spur output pinion drives two gears; one on each of two traction drums which rotate on three anti-friction bearings would be (.86 X :89 X .95) X 100% = 69.1%.

The above factors are as follows:

.86 is the overall hydraulic pump efficiency factor
.89 is the overall hydraulic motor efficiency factor
.95 is the planetary gearbox efficiency factor
.95 = (1-.02 - .01 X 3) is the efficiency factor for the final gear mesh and the three drum bearings

If 130 HP is required at the output of the winch, the input power required would be 130/.691 = 188.1 HP, so if an electric motor is to be used, it would have to be sized at 200 HP which is the next larger size available.



PHOTOGRAPH 5

CHAPTER 10

Rope and Cable Levelwinding at the Winch

REYNOLD SMITH

1.0	INTRODUCTION	10–2
2.0	THE LEBUS SHELL	10–3
	2.1 GROOVING PATTERN	10–3
	2.2 SOFT CROSSOVER SPOOLING	10–6
3.0	DESIGN CONSIDERATIONS	10–7
	3.1 CABLE DIAMETER	10–7
	3.2 DRUM CONSTRUCTION	10–7
	3.3 LEBUS SHELL MATERIALS	10–11
	3.4 WINCH DRUM FILLET WELD	10–11
	3.5 DRUM CORE SIZE	10–12
	3.6 HELIX CONSIDERATIONS	10–12
4.0	LEVEL WIND CONSIDERATIONS	10–12
	4.1 LEVEL WIND TYPES	10–14
	4.2 DIAMOND SCREW DRIVES	10–14
	4.3 AUTOMATIC FLEET ANGLE COMPENSATOR	10–16
	4.4 SINGLE LAYER SPOOLING	10–18
5.0	SPECIFICATIONS FOR GROOVED SHELLS AND COMPENSATORS	10–18
	5.1 GROOVED SHELL	10–18
	5.2 FLEET ANGLE COMPENSATOR	10–20

1.0 INTRODUCTION

This chapter will deal with the design of cable handling systems from the standpoint of achieving the most efficient cable spooling possible. Primarily the multiple layer spooling of oceanographic cables is concerned with not only the deployment and retrival of the cable, but also the prolonging of its working life through proper handling. To many users the term "Level Winding," is simply the distribution of a wire or cable between the flanges of a winch drum in such a way that the wire neither piles up in the middle of the drum nor creates low spots at the drum flanges.

For the purposes of this discussion, perhaps a better term to describe the cable handling process would be "Controlled Spooling." In a controlled spooling situation the cable handling system is designed so that each wrap of each layer of cable is at some pre-determined place on the winch drum each time the cable is spooled onto the drum. Controlled spooling will also cause each wrap to lie adjacent to the preceeding wrap leaving no voids for succeeding wraps to fall into causing uneven spooling and will result in each layer of cable having exactly the same number of wraps as the preceeding one.

In order to clarify the terminology used in this discussion, a few basic terms need to be defined at this point. The commonly used terms in this chapter are as follows:

1. Wrap: A wrap of cable or rope is defined as one complete turn of the wire around the winch drum.

2. Dead Wrap: This refers to the wraps of cable on the winch drum which remain on the drum at all times.

3. Active Wraps: This terms applies to the wraps or cable which are actually wound and unwound from the winch drum during actual at-sea operations.

4. Layer: This term, also referred to as a "fleet," is made up of the number of wraps required to fill the space between the drum flanges in a single pass.

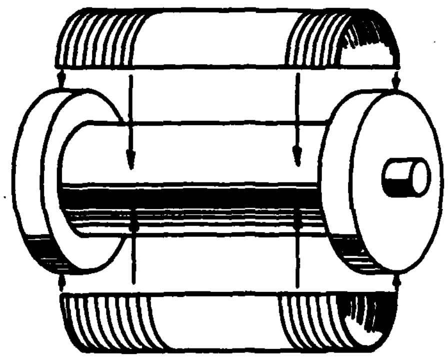
In order to achieve a controlled spooling situation, a number of factors must be taken into account during the design of the system. Included in these areas of concern are the Lebus* shell configuration, winch drum dimensions, fleet angles, and level wind gearing. The remainder of this chapter will deal

* Lebus is a registered trademark of Lebus International Inc.

with the specifics of obtaining controlled spooling situations at the winch.

2.0 THE LEBUS SHELL

The Lebus shell is essentially a grooved cylinder, manufactured from either steel, aluminum, or fiberglass, which is designed to assure the proper seating of a specific rope or cable and the proper movement and spacing of that wire between the flanges of the winch drum. In order to effectively use the Lebus shells the winch drum must have flanges which are perpendicular to the barrel or core of the smooth winch drum. The shells, when delivered, are split for easy installation on the winch or take-up spool and attachment can be accomplished by either welding or bolting the shell in place (Figure 10-1). Where winch systems utilize more than one size of wire or cable during their operational life, it is recommended that the bolt-on technique be used.

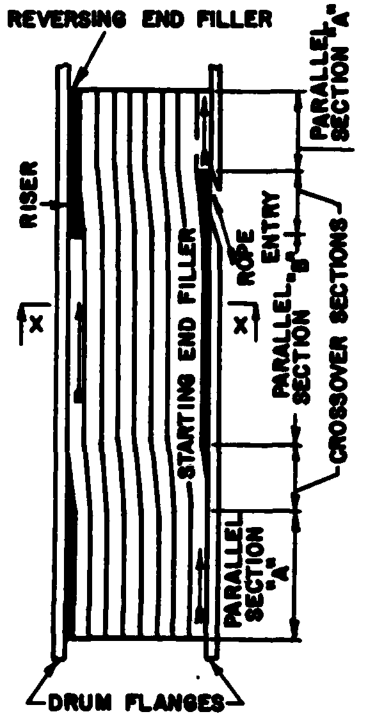


LE BUS SHELL CONFIGURATION
FIGURE 10-1

2.1 Grooving Patterns

In order to obtain a controlled multi-layer spooling of the rope or cable it is necessary to establish a pattern for the first layer that will be repeated by the second, third, etc. The precise placement of the wire is the function of the groove pattern on the Lebus shell. The groove pattern is designed for a specific rope or cable and for installation on a particular winch. It is important when ordering the shells that adequate data be provided to the company in order to assure a perfect fit and proper operation. Ordering details are provided later in the text.

The groove pattern (Figure 10-2) controls the area where one layer rises to meet the next on the drum and continues to maintain an even spooling of the wire regardless of the number of layers involved or the speed of the winch.

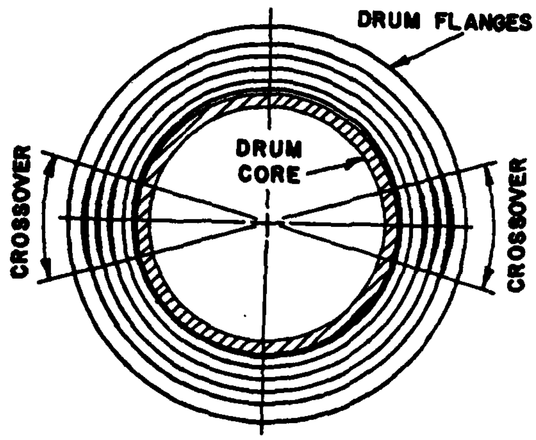


LE BUS SHELL GROOVE PATTERN
FIGURE 10-2

The initial pattern set by the grooves on the shell controls the point at which one layer crosses over the layer below (Figure 10-3). The layer, thus applied, will alternate between a right hand helix and a left hand helix at the crossover points. The crossovers that occur are thereby counterbalanced and the drum remains in a state of balance in its rotation when spooling multiple layers of wire.

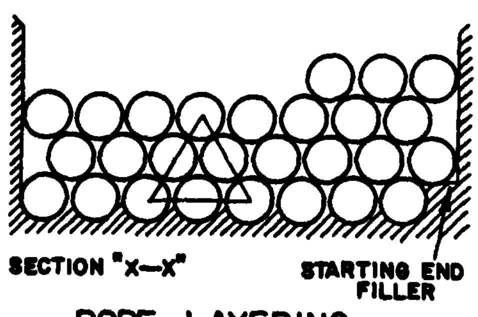
In order to accomplish the perfect multi-layer spooling of the cable on the winch drum, the Lebus system is presently the only product on the market that eliminates the 360 degree continuous cross winding of the cable as found on smooth drum winches. This system with its groove pattern reduces the cross winding to approximately 20% of the circumference of the drum while 80% of the wraps remain parallel with the flanges of

the drum. In view of this patterning, each layer of wire then becomes the groove pattern for each successive layer.



CROSSOVER SECTIONS
FIGURE 10-3

While the initial groove pattern establishes the layering of the wire it also succeeds in creating a cross section pyramid pattern in the wire layers (Figure 10-4). The pyramid pattern created by spooling wire in this manner lessens the stresses placed on the winch drum flanges as compared to a smooth drum winch. Also the pyramid pattern minimzies the scrubbing and chopping of the cable during winding and unwinding.



ROPE LAYERING
FIGURE 10-4

2.2 Soft Crossover Spooling

The soft crossover in the shell design reduces the wire rise allowing a gentle action at the crossover point which is the point of shock for the wire and also serves as a shock absorber. By reducing the crossover rise or hump found in all situations where wire is multi-wrapped, the user receives a much smoother action on the drum rotation thereby eliminating the out of round looping movement usually seen at crossover points.

By utilizing this soft crossover spooling the horizontal scrubbing and abrasion of the wire as it starts its movement across the drum is reduced. Vertical abrasion is also reduced since the use of this approach also reduces the hump or rise at the crossover section. A maximum wire life can be expected from the use of soft crossover spooling when combined with proper wire care and maintenance.

Figure 10-5 illustrates a side view of a drum core and depicts what is meant by "Soft Crossover Spooling." It will be noticed in the crossover sections, which are 180° apart, that the flat or level area over which the wire will pass is used to create an area of limited rise. By flattening this area, wire build-up is reduced to a minimum as the wire moves between the flanges of the winch drum. This can be compared with Figure 10-3 which shows the normal crossover area found in the Lebus shell.

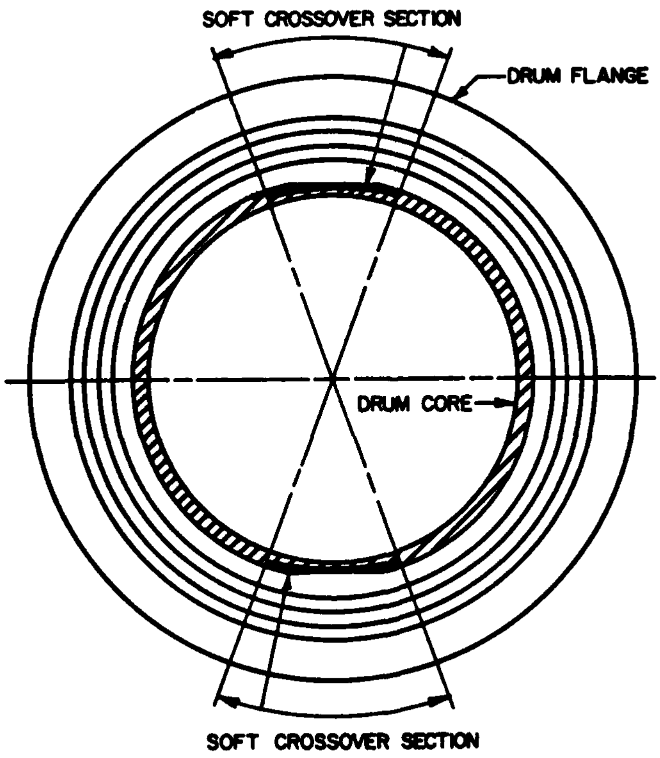


FIGURE 10-5

3.0 DESIGN CONSIDERATIONS

As mentioned earlier, there is a significant difference between a level-wound wire and one which is installed on the drum under controlled spooling conditions. Photograph 1 illustrates a traditionally level-wound wire rope while Photograph 2 illustrates the same type of wire wound on under controlled spooling conditions. In order to achieve the more desirable controlled spooling, a number of design considerations must be understood.

3.1 Cable Diameter

Probably the most important factor in achieving controlled spooling on a Lebus shell is a cable of known diameter which is insistant throughout its entire length. This diameter should remain constant under working loads, for a reduction in diameter of only 4% or 5% will result in a loss of the spooling pattern. In this regard prestressed ropes and cables are preferred. The use of unstressed wire can result in uneven spooling, through no fault of the installed equipment.

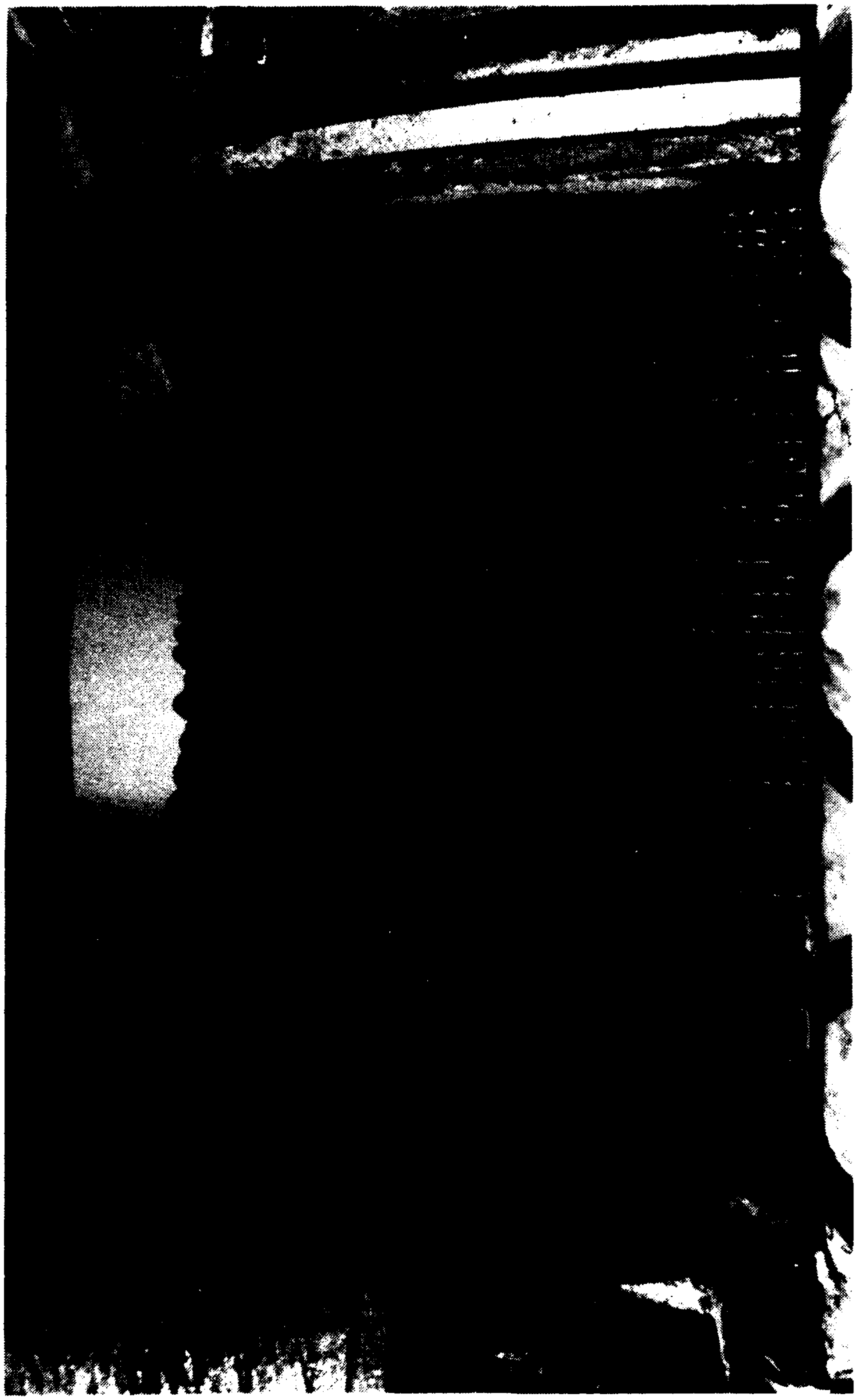
Lebus shells have been manufactured for cables as small as .090 inches and as large as 4 inches in diameter, all of which have been evenly spooled using the Lebus System.

Generally, the construction of a rope or cable is not critical to the Lebus shell as long as the diameter is held constant. Obviously, the more round the wire is the better it can be spooled onto the drum and the more even the finished layer. Three stranded oceanogrpahic wire rope presents some problems due to its construction, but has been successfully controlled spooled to a depth of sixteen layers.

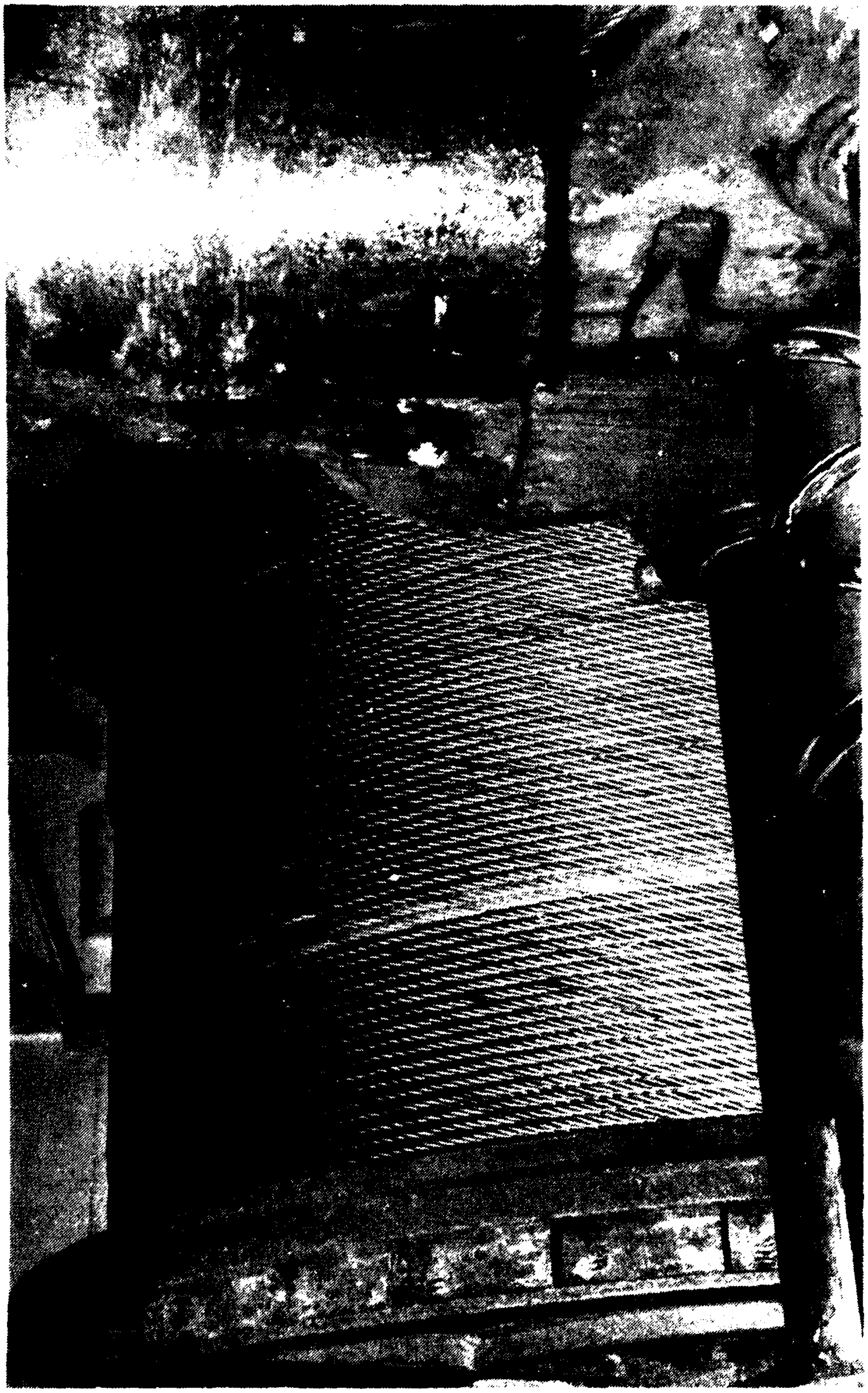
3.2 Drum Construction

Just as important to controlled spooling as the wire diameter is the width between the flanges of the winch drum. In general, the tolerance of a drum width must not exceed plus or minus 3% of the cable diamter if accurate spooling is to be achieved. Flange widths which fall outside of this tolerance range will not allow the establishment of a proper groove pattern and will result in poor wire spooling.

Of equal importance is the construciton of the drum itself and a consideration of tne stresses placed on it by the wire. Many winch drums have failed in service due to insufficient knowledge of inherent drum stresses. Two primary causes of drum failures which users should be aware of are as follows. A failure of the drum core due to extreme winding tensions which exceed the ultimate strength of the core can result in an oblongation of the core. Although an uncommon mode of failure, the drum core should occasionally be checked for roundness.



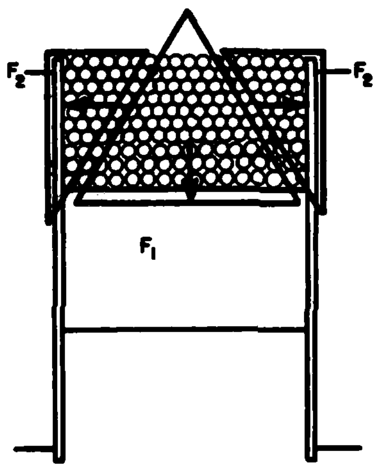
PHOTOGRAPH I



PHOTOGRAPH 2

The second drum failure mode which is most common is a flange failure. Pressure on the drum flanges at the root or fillet of the flange area and the reinforcement from the barrel to the height of the spooling pattern can be tremendous. The stresses responsible for this type of failure usually occur when forces are imposed by a coil wedging between the drum flange and an adjacent coil, particuarly when a bad pick up occurs at the same time. In addition, high stresses can be exerted due to the terminal expansion of an oceanographic wire which is brought up from near 0°C conditions and stirred tightly at high surface temperature.

A good distribution of the loads induced by wire which is spooled in multiple layers can be achieved through the use of the Lebus shell. Figure 10-6 illustrates the force distribution exerted on the drum by pyramidal build up of wire achieved with a shell. Even with this distribution of forces, damage to substandard drums is still possible and drum dimensions of questionable equipment should be carefully monitored or replaced.



FORCE DIAGRAM SHOWING LOAD EXERTION ON DRUM FLANGES DUE TO WIRE LINE

FIGURE 10-6

If Lebus shells are contemplated for a winch which has previously been operated with a smooth drum, the user should work in close cooperation with the shell manufacturer in order to achieve the desired result. For example, if the drum in question will accommodate 50 wraps or more of wire per layer, several groove pitches can be calculated in 1% increments for any actual drum width. Of these at least one will provide a workable pitch for the specific application. Conversely, if there are less than 50 wraps per layer, the actual drum width that will be used must be agreed upon by both the user and the manufacturer of the grooved sleeve. Under these conditions it may be necessary to add spacer plates to the flanges in order to

arrive at a suitable drum width.

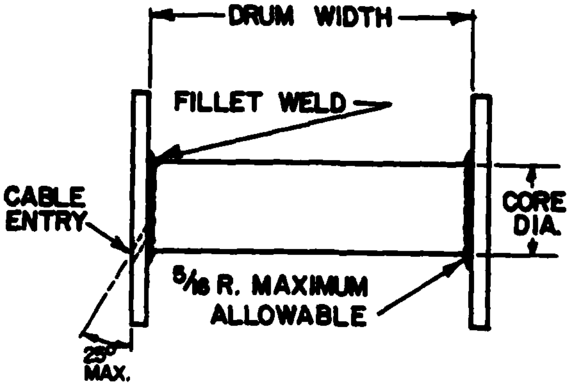
Other considerations to be taken into account when fitting a Lebus shell to a winch drum is the core dameter. This dimension is not as critical as the drum width, but still should be held to some practical machine tolerance. In most cases a specification of plus 1/16 inch minus zero will be adequate.

3.3 Lebus Shell Materials

Since these shells represent a unique winch component that is specifically designed for a particular usage where long life is required, the first choice in material is a mild steel. Shells can also be furnished in stainless steel, aluminum, and several types of plastic where required. In some cases where a large quantity of identical sleeves are required, they can be fabricated out of fiberglass. The choice of materials is wholly, dependent upon the individual application and the manufacturers expertise should be relied upon to make the material selection.

3.4 Winch Drum Fillet Weld

The size of the fillet weld that occurs where the drum flange and the core are joined is a minor, but important factor in the proper matching of the Lebus shell to the winch drum. For a normal shell installation, this weld (Figure 10-7) should not exceed a 5/16 inch radius. The actual sizes of this weld should be supplied to the manufacturer at the time the shell is ordered.



WINCH DRUM FILLET WELD

FIGURE 10-7

3.5 Drum Core Size

Lebus shells have been produced for drum cores as small as three inches in diameter and as large as eighteen feet in diameter. Obviously, the larger the drum core, the less the unit pressure a wrap exerts on the groove surface or the wire below. This unit pressure can be easily determined using the following formula:

$$\text{Unit Pressure} = \frac{2 \times \text{Cable Tension}}{\text{Core Diameter} \times \text{Cable Diameter}}$$

3.6 Helix Considerations

Grooved shells can be produced with either a right-hand helix or a left-hand helix at the crossover areas depending on individual system requirement. When ordering a shell, it is important that an accurate drawing be supplied which details the cable entry from the flange. The success of the controlled spooling of the wire depends upon the exact matching of the wire entry point through the flange and its entry into the shell's groove pattern.

The decision of whether to employ a right-hand or left-hand groove depends on whether the drum is overwound, and which flange is most convenient for terminating the shipboard end of the wire. The lay of the wire, either right or left hand, is usually ignored when multilayer spooling using a Lebus shell. This is possible since the groove pattern properly spaces the wire and the various layers alternate between right- and left-hand helixes in the crossover areas.

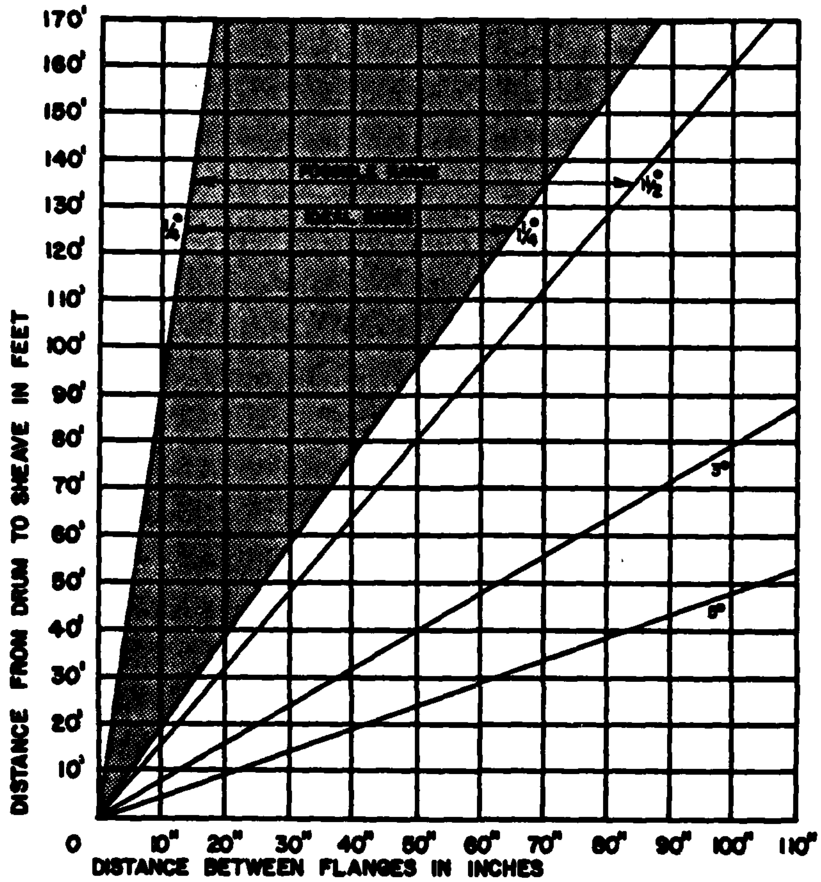
4.0 LEVEL WIND CONSIDERATIONS

In order to obtain a controlled multi-layer spooling with a properly designed Lebus shell several conditions must be satisfied. First the fleet angle must not exceed 1.25 degrees or a ratio of two feet of distance to the fixed point for every one inch of drum width (Figure 10-8). This ratio assumes that the fixed point is on a line centered between the winch drum flanges. The achievable fleet angle for a given situation is a major consideration in the design of the drum width. However, when the fleet angle cannot be held at 1.25 degrees or less it is necessary to use some type of fleet angle compensator.

The second consideration in a multi-layer spooling situation is the importance of sufficient cable tension to hold the wraps in place on the winch drum. Insufficient cable tension will result in uneven loading of the drum and can lead to premature wire abrasion and uneven spooling. Generally the wire should be spooled under a tension of at least ten to fifteen percent of the anticipated working tension.

ASSUME SHEAVE IS LOCATED ON CENTERLINE OF DRUM

SHADED AREA IS THE PROPER RANGE OF FLEET ANGLE



NOTE: FOR ANY FLEET ANGLE BETWEEN 0° AND 1/4°, OR BETWEEN 1-1/4° AND 1-1/2°, PLEASE CONTACT LEBUS ENGINEERS FOR THEIR RECOMMENDATIONS

FLEET ANGLE CHART
FIGURE 10-8

4.1 Level Wind Types

As was mentioned earlier, if the fleet angle exceeds 1.25 degrees some form of fleet angle compensation must be used if controlled spooling is to be achieved. There are several types of level winding mechanisms in use today which can be classified in three categories: those that must be geared to the drum shaft; those which depend on the tension in the cable for proper operation; and a third type which is operated by some independent force. The two most popular types used with the Lebus shell are the diamond screw and the Lebus Automatic Fleet Angle Compensator.

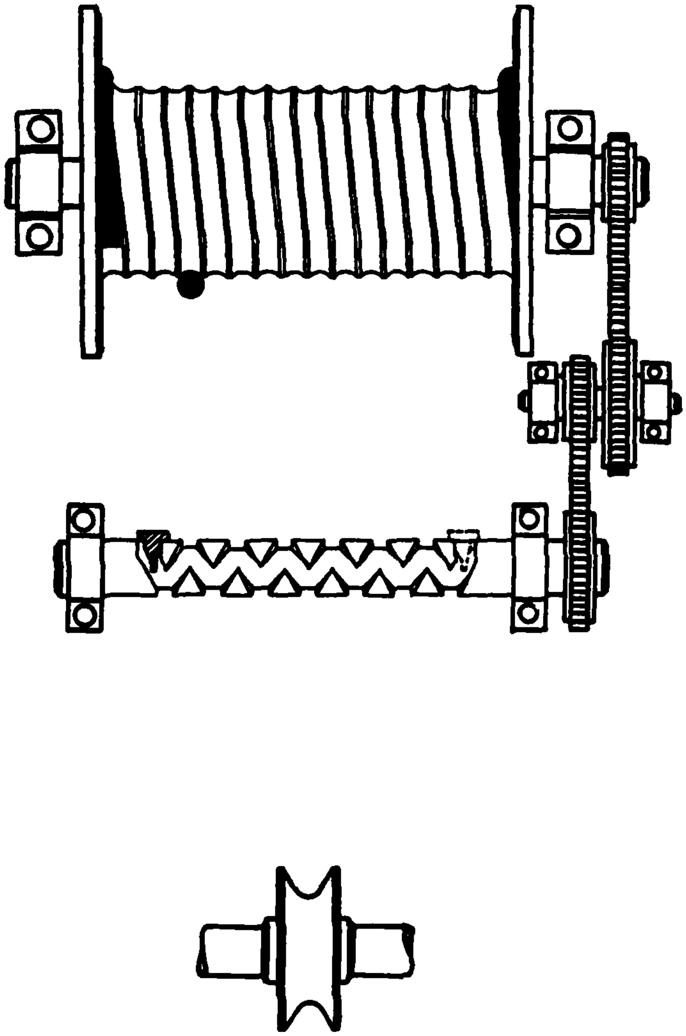
4.2 Diamond Screw Drive

The diamond screw type of level wind has a right- and left-hand thread blended together at the ends of the screw to form a turnaround area for a matching pawl (Figure 10-9). The screw is driven by the winch drum's rotation with the pawl traveling in the thread grooves which in turn drives the fairlead back and forth between the flanges (Photograph 2). The gear ratio required between the winch drum and the diamond screw is a function of the number of wraps of cable in each layer and the number of revolutions the screw makes to cycle the fairlead from one turn around to another.

It is important to remember that on winches whicn use more than one size wire or cable that a change in wire size results in a change in the number of wraps occurring on the drum. When this is the case not only the shell must be changed, but also the winch drum/diamond screw ratio of controlled spooling of the different wires is to be achieved. A variable ratio is not recommended for these conditions, but instead a changable positive gear or sprocket ratio will result in the fairlead cycling in exact unison with the different sized cable spooling on and off the drum.

The length of travel of the fairlead should create a fleet angle of approximately 3/4 degree at each flange. The length of travel of the fairlead along the diamond screw also includes a critical period at each turnaround point called the dwell time. This is the period of time required for the wire to rise from the just completed layer and make the first wrap of the next layer. During this period the fairlead must "Dwell" or remain stationary at the end of the diamond screw to allow this first wrap to be laid in place before traversing the screw in the other direction.

It is desirable to have the cable go from the drum to the fairlead and out to some fixed point such as a flag block before entering the sheave train. This condition reduces the end thrust on the diamond screw thereby reducing the size of the screw required to operate the fairlead.



DIAMOND SCREW CONFIGURATION
FIGURE 10-9

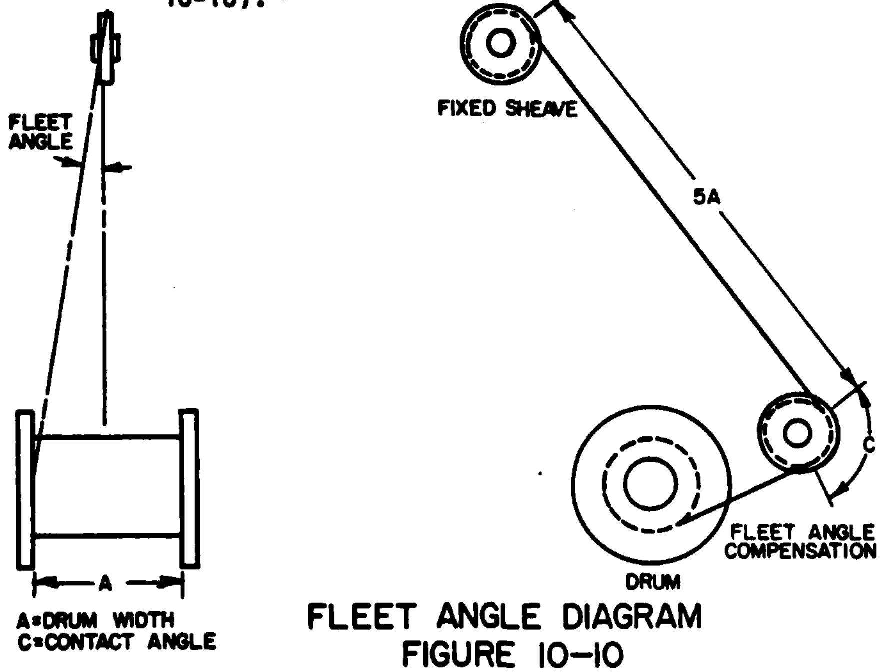
The diamond screw mechanism probably has as much design work involved in it as the rest of the winch design. The adaptation of a diamond screw to an existing winch requires that the winch have an exposed drum shaft that is keyed to the drum in order to mount the gear or sprocket used to drive the screw. Hydraulic winches with no exposed drum shaft are very difficult, if not impossible to modify for a diamond screw level wind.

4.3 Automatic Fleet Angle Compensator

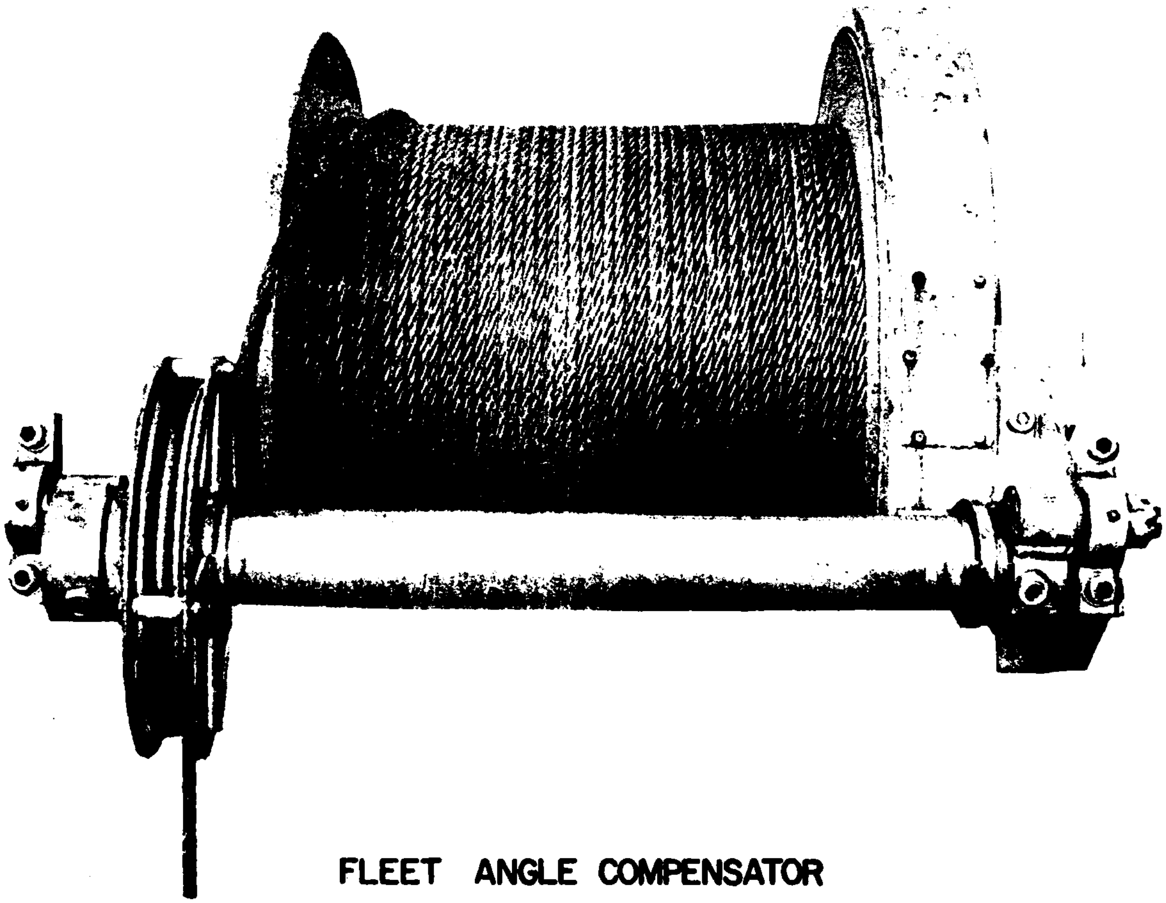
The Lebus Automatic Fleet Angle Compensator is basically an ordinary fleeting sheave which is free to rotate and slide along a shaft between eccentric centers (Photograph 3). The movement of the sheave along the shaft is entirely contolled by the wire spooling onto the drum. The tension of the wire causes the shaft to rotate on an eccentric so that the sheave moves in the path of an arc allowing the fleet angle to remain constant. During this process the wire is fed onto the drum in a parallel to the flange movement even at the largest angle from the fixed sheave.

Although there are no direct mechanical connections between the winch drums and the compensator apart from the wire itself it still requires that the unit be adjusted for the wire in use. For the fleet angle compensator to function properly the following conditions must be met:

1. The cable must go from the drum over the compensator sheave to a fixed point such as a fairlead or fixed sheave.
2. To avoid excessive wire angles the minimum distance between the fixed point and the compensator sheave must be at least five times the drum width (Figure 10-10).



FLEET ANGLE DIAGRAM
FIGURE 10-10



FLEET ANGLE COMPENSATOR

PHOTOGRAPH 3

3. The wire must have a minimum angle of contact with the compensator sheave of at least 60 degrees.

4. The drum should be equipped with either a grooved shell for multi-layer spooling or a helical groove for single layer spooling.

5. Sufficient wire tension must be maintained during spooling to ensure proper placement of the wire.

If these minimum requiremnts cannot be met due to vessel limitations, then it is unlikely that perfectly controlled spooling can ever be achieved. In extreme cases it may be wiser to install the diamond screw level wind instead of the fleet angle compensator. The differences in the end spooling product are depicted in Photographs 1 and 2 in this chapter.

4.4 Single Layer Spooling

Up to this point our discussions have dealt with the controlled spooling of multi-layers of wire or cable. However, in cases where all the wire required can be spooled in a single layer a helical groove machined into the drum is recommended. In this application the machined grooves should be arranged so that the pitch of the groove prevents the individual wraps of wire from touching one another.

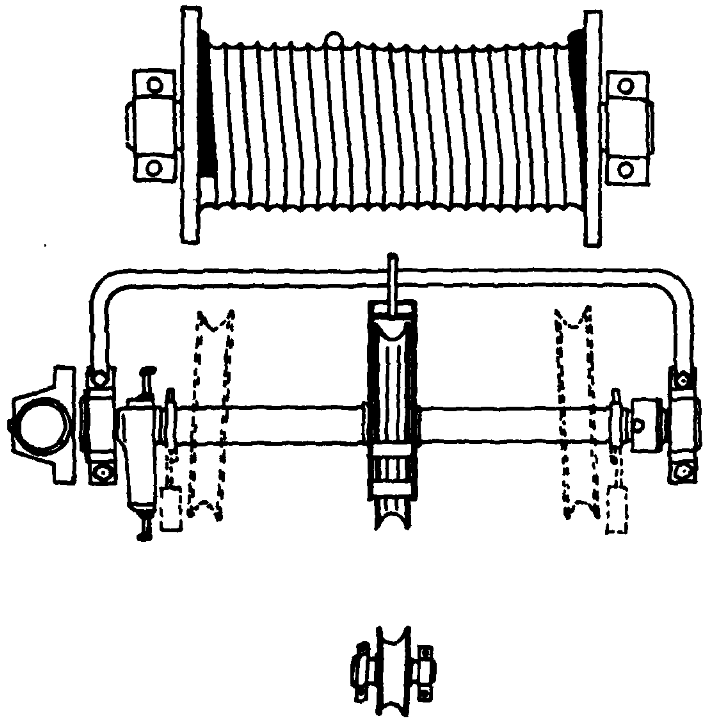
The grooving on a single layer drum can be provided in a variety of helical patterns and configurations designed to meet individual needs. For exmaple, a helical grooved drum can be made with more than one groove pitch in order to accommodate an increase in cable size as occurs when cables are paired for only a portion of their length. Also the groove pitch can be increased to allow for more fleet angle at one end of the drum than the other. For the purposes of single layer spooling fleet angles of 2.5 to 3 degrees can be easily accommodated.

5.0 SPECIFICAITONS FOR GROOVED SHELLS AND COMPENSATORS

As has been mentioned in the text, it is crucial that good communications exist between the potential users of these devices and the manufaturer if a satisfactory end product is to be achieved. The following sections outline the information which should be made available to the manfuacturer if this equipment is considered for use.

5.1 Grooved Shell

The following data and Figure 10-11 can be used to establish a point at which a grooved shell can be produced for a specific application. Specific items of interest in the case are as follows:



1. Winch drum diameter.

2. Winch drum width or length.

3. Construction and size of wire rope (Sample should be provided if possible).

4. Amount of wire on drum.

5. Are drum flanges worn?

6. O.D. of drum flanges.

7. Wire Rope Speed.

8. Distance to fixed sheave.

9. Size of drum fillet.

10. Is drum grooved or plain?

11. Wind used for oceanography, crane, tec.

12. Type of material from which drum and flanges are made

    a) Steel
    b) Cast iron
    c) Other

13. Maximum and minimum load on rope while spooling.

14. Specify right- or left-hand grooving to indicate correct wire entry to drum and movement.

15. Wire entry onto the core, through flange or core.

5.2 Fleet Angle Compensator

The following items should be answered when considering the incorporation of a compensation device.

1. Elevational view of winch or hoisting equipment.

2. Distance to first fixed sheave from the drum.

3. Drum dimensions.

4. Wire or cable size.

5. Are there any slack rope conditions involved in equipment operation?

6. Approximate weight of hook or block.

7. Minimum or maximum loads to be hoisted.

8. Wire or cable speed.

9. Can compensator be mounted in front, on top or in back of the winch drum?

10. Is present drum grooved?

CHAPTER 11

Winch Hydraulic Systems

RAYMOND BOUDREAUX

1.0	INTRODUCTION	11-3
2.0	DESIGN AND OPERATION OF HYDRAULIC SYSTEMS	11-3
	2.1 FUNDAMENTALS OF HYDRAULIC SYSTEMS	11-3
	2.2 OPEN LOOP CIRCUITS	11-5
	2.3 CLOSED LOOP "TRANSMISSION" CIRCUITS	11-5
3.0	HYDRAULIC CIRCUITS	11-7
	3.1 INSTALLATION AND PLUMBING	11-8
	3.2 COMPLEMENTARY COMPONENTS	11-8
4.0	HYDRAULIC FLUIDS (PETROLEUM & FIRE-RESISTANT TYPES)	11-10
	4.1 FLUIDS VOCABULARY	11-13
	4.2 PETROLEUM FLUIDS	11-14
	4.3 FIRE-RESISTANT FLUIDS	11-14
	4.4 SYNTHETIC FLUIDS	11-14
	4.5 WATER-GLYCOL FLUIDS	11-15
	4.6 ASTM VISCOSITY CLASSIFICATION FOR INDUSTRIAL SYS.	11-15
5.0	CLEANLINESS OF HYDRAULIC SYSTEMS	11-17
	5.1 STORAGE AND HANDLING	11-17
	5.2 FLUIDS IN SERVICE	11-17
	5.3 GRAVITY SETTLING	11-18
	5.4 CENTRIFUGING	11-19
	5.5 CONTINUOUS RECONDITIONING	11-19
	5.6 FILTRATION FOR FLUID POWER SYSTEMS	11-20
	5.7 FILTER SELECTION FACTORS	11-21
6.0	HYDRAULIC SYSTEM MAINTENANCE	11-23
	6.1 DRAIN SCHEDULES	11-24
	6.2 CLEANING AND FLUSHING	11-25

7.0 TROUBLE SHOOTING HYDRAULIC SYSTEMS 11—29
7.1 IMPROPER OPERATION OF PUMP 11—29
7.2 IMPROPER OPERATION OF ACTUATING MECHANISMS 11—33
7.3 IMPROPER OPERATION OF ACCUMULATOR 11—34
7.4 EXCESSIVE HEATING OF OIL IN SYSTEM 11—35
7.5 IMPROPER OPERATION OF FLUID MOTORS 11—36

1.0 INTRODUCTION

Most hydrographic and oceanographic winches in use today utilize some form of hydraulic circuits in order to operate effectively. Although different types of power transmission systems are available, i.e., AC, DC and direct mechanical drives, the electro-hydraulic option is frequently selected for a variety of reasons. Since a hydraulic system offers a relatively simple method of applying both large and small forces with a minimum of effort, and its associated control systems can be developed in a wide variety of configurations, the flexibility of a properly designed system is astounding.

Hydraulic equipment, as it is available today, is so highly versatile that it can provide, with ease, several interlocked and coordinated options or operations which would be difficult to obtain by other means. Hydraulic power because of its flexibility, is distinctive and most certainly has a place in the oceanographic community where, frequently, a wide variety of functions must be performed simultaneously aboard ship. Frequently, hydraulics will be used to power the winches, A-frames, and lift equipment aboard the vessel as well as being used to power various pieces of lift equipment. The discussions in this chapter will treat hydraulics as they apply to oceanogrpahic applications and will deal with the design, cleanliness, maintenance, noise, and the trouble shooting of a hydraulic system.

2.0 DESIGN AND OPERATION OF HYDRAULIC SYSTEMS

Any attempt to completely cover all aspects of winch hydraulic systems design and operation would require a rather lengthy book. Control systems, valving, different types of pumps and motors, as well as special design cylinders vary considerably within the oceanographic community and for these reasons, this discussion will be limited to information drawn from sources that can be applied in general.

2.1 Fundamentals of Hydraulic Systems

Modern hydraulic systems on many types of equipment seem complicated, and some are. Actually, however, their fundamental design is quite simple. This can be illustrated by building up piece by piece a hypothetical hydraulic system.

In about 1650 Blaise' Pascal discovered the funamental law of physics upon which all modern hydraulic systems are based. Essentially, Pascal's law states, "Pressure exerted on a confined liquid is transmitted undiminished in all directions and acts with equal force on all equal areas."

Since liquids are essentially incompressible and are fluid, i.e., flow rapidly, the pressure applied to the liquid at one point will be transmitted to any point the liquid reaches.

Hydraulic systems are assemblies of units capable of accomplishing this, for they contain fundamentally a unit for generating force (i.e., pumps), suitable tubes or pipes for containing and transmitting the fluid under pressure, and units in which the energy in the fluid will be converted to mechanical work (cylinders and fluid motors). Of course, any operative system must contain, in addition, such auxiliary equipment as valves to control and direct the flow of fluid and limit the pressure, a reservoir to contain a supply of fluid, and return lines to carry the fluid back to the reservoir after it has served its purpose. In addition, many modern hydraulic systems contain other auxiliary equipment which will be described in chapters to follow.

Pascal's Law is illustrated in Figure 11-1. If a force of 10 pounds is applied on the piston in Cylinder A having an area of 1 sq. in. then the pressure created on the oil just below the piston in pounds per square inch will be 10 lbs. divided by 1 sq. in. or 10 pounds per square inch. Since Pascal's Law states "this pressure will be transmitted undiminished in all directions and acts with equal force on all equal areas," then, obviously, the pressure created in Cylinder B and on the piston in this cylinder will be 10 pounds per square inch. If the piston in Cylinder B has an area of 10 square inches, then the total force against this piston is 10 lbs. per sq. in. x 10 sq. in. or 100 pounds.

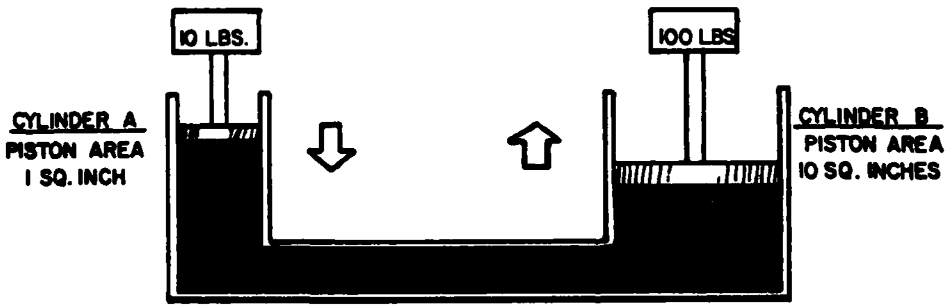


FIGURE 11-1 PASCAL LAW

Thus, a small force applied to a small piston can be transformed into a much greater force acting on a larger piston. However, the movement of the larger piston through a given distance requires that the smaller piston be moved a greater distance. For example, in Figure 1, if the piston in Cylinder A were moved down ten inches, then the volume of oil displaced would be 10 inches times 1 square inch (the area of the piston) or 10 cubic inches. Since fluids are essentially incompressible, the displaced fluid in Cylinder A would have to go into Cylinder B. To find the distance this piston would be moved

up, simply divide 10 cubic inches by the area of this piston (10 square inches), and the distance will be found to be 1 inch. Thus, in the above example a movement of the piston in Cylinder A through a specific distance and caused by a small force will result in a greater force on the piston in Cylinder B, but the distance the latter travels will be much shorter.

2.2 Open Loop Circuits

The basic principles of hydraulics as defined in Pascals Law can be applied in many ways to transmit power. A hydraulic pump could replace Cylinder A and a hydraulic cylinder or motor would replace Cylinder B in Figure 11-1. By varying the volume and pressure of the system, speed and force can be increased or decreased as required.

As applied to winch hydraulic systems, two types of circuits are used -- open loop system and closed loop systems. The open loop system shown in Figure 11-2 is most often used to operate hydraulic cylindes or motors for crane booms, "A-frames," capstans, steering systems, and other devices where accurate speed variation is not necessary. As an example a hydraulic motor can be used in place of the cylinder shown to achieve a desired operation. The open loop system is generally used for start-stop and reversing operation.

The open loop system generally consists of a reservoir, pump, relief valve or unloading valve, directional control valve, and cylinder or hydraulic motor. Filters are used as required in the reservoir on the suction line, and in the return line in a suitable location. Filters, in this and other hydraulic systems, are critical to the efficient operation of the system as free floating dirt or metal particles will result in unexpected system failure.

The pumps usually used in these systems are either a gear or valve type and only occasionally are piston pumps used in the open loop ciruit. Where higher speeds and pressures are desired multiple pumps and motors are frequently employed. Because of the nature of open loop circuits, i.e., start-stop motions, the pumps and motors used are generally of the fixed displacement type.

2.3 Closed Loop "Transmission" Circuits

In order to provide a more versatile power transmission system with variable speed control, closed loop hydraulic circuits have been develped. Within this system the main pump, or "high pressure" loop, is isolated from the rest of the system. Hydraulic oil is circulated between the main pump and motor only and all valving is usually built into the pump and motor. A second pump is used to draw oil from the reservoir and circulate it for cooling, lubrication, and replenishing of the system.

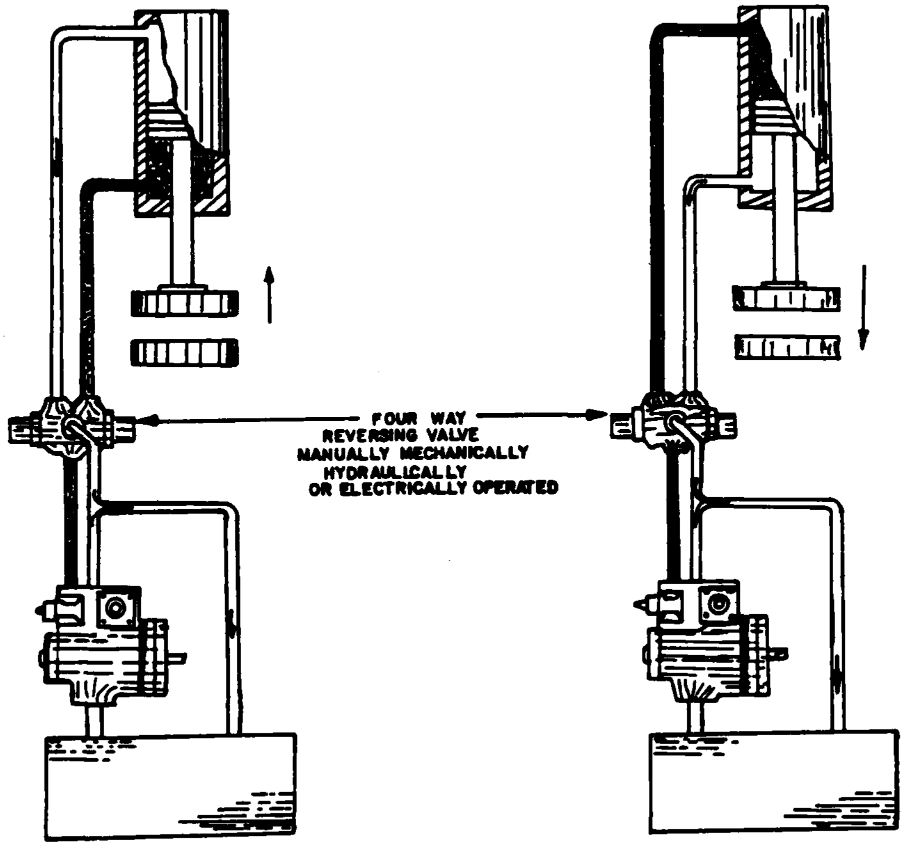


FIGURE 11-2

Double acting cylinder (thrust in both directions). The piston in this double acting cylinder can be raised or lowered at will under controlled conditions.

This second pump system could be classified as an "open loop" system, and is sometimes used to operate brakes, clutches, or cylinders. These closed loop transmission systems are sometimes referred to as hydrostatic transmissions.

The most common closed loop systems use a variable displacement axial piston pump and motor with a gear pump in the open loop circuit.

3.0 HYDRAULIC CIRCUITS

The hydrostatic transmission offers infinite control of speed and direction. The operator has complete control of the system with one lever for starting, stopping, forward motion or reverse motion. Automatic control systems are also available for automatic torque, speed, and horsepower operation.

Control of the variable displacement, axial piston pump is the key to controlling the machine. Prime mover horsepower is transmitted to the pump. When the operator moves the control lever, the swashplate in the pump is tilted from neutral.

When the variable pump swashplate is tilted, a positive stroke to the pistons is created. This, in turn, at any given input speed, produces a certain flow from the pump. This flow is transferred through high pressure lines to the motor. The ratio of the volume of flow from the pump to the displacement of the motor will determine the speed of the motor output shaft. Moving the control lever to the opposite side of neutral, the flow from the pump is reversed and the motor output shaft turns in the opposite direction. Speed of the output shaft is controlled by adjusting the displacement (flow) of the transmission. Load (working pressure) is determined by the external conditions, and this establishes the demand on the system.

Pump and motors are contained in separate housings or may be connected by a common end cap. All valves required for a closed loop circuit are included in either the pump or motor assemblies. A reservoir, filter, cooler and lines complete the circuit.

Figure 11-3 illustrates the general appearance of the components of a heavy duty transmission. Different manufacturers supply transmissions that vary somewhat from the illustration, but the basic operating principles are the same.

3.1 Installation and Plumbing

The system in which the hydrostatic pump and motor is operated should provide an environment compatible with the requirements of the transmission. The requirements of the complementary components necessary to complete the hydraulic circuitry are described below. The arrangement of the components and their respective sizes are shown on Figure 11-3 and Plumbing Reference Chart, Figure 11-4.

3.2 Complementary Components

Reservoir: (Figure 11-3, Item 1)

A suggested minimum reservoir volume (in Gallons) is five-eighths of the total charge pump flow per minute (in GPM) with a minimum fluid volume equal to one-half charge pump flow. This minimum reservoir volume will provide for a minimum of 30 seconds fluid dwell at the maximum reservoir return flow in the system.

The outlet port to the charge pump inlet must be positioned above the bottom to take advantage of gravity separation

and prevent any large foreign particles from entering the suction line. A 100 mesh screen is recommended over the outlet port to further assist large particle separation. The fluid level in the reservoir must always be above the outlet port.

The reservoir inlet (fluid return) should be positioned in such a way that return flow is directed into the interior of the reservoir to provide for maximum dwell and most efficient de-aeration of the fluid.

A drain in the reservoir is recommended which would permit a complete fluid change without disconnecting other normal hydraulic connections. This would also provide a water drain and permit flushing in the event of excess system or component contamination.

A filler port (Figure 11-3, Item 12) should be provided that minimizes the potential for contamination entering the system during servicing or during operation. A closed reservoir is recommended to reduce introduction of contamination and be designed so that the recommended charge pump inlet pressure and case drain presssures are not exceeded.

Reservoir Shut-Off Valve: (Figure 11-3, Item 2)

It is recommended that a shut-off valve be installed between the reservoir outlet and filter inlet, on those systems incorporating a filter installed outside the reservoir to facilitate a filter change without a large loss of fluid and to minimize system contamination.

The minimum inside diameter of the valve should be equal to or greater than the inside diameter of the charge pump inlet line so that the recommended charge pump inlet pressures are not exceeded.

Filter: (Figure 11-3, Item 3)

The fluid supplied to the charge pump system can be filtered by a good quality 10 micron nominal rated suction filter and should not incorporate a by-pass valve. This type of filtration system will give the greatest degree of reliability in keeping the system free of contamination. Filter clogging causing reduced inlet pressure to the pump beyond limits specified will eventually result in reduced transmission control response. The transmission will become slow and sluggish. This will occur before any damage to the transmission results and provide ample indication that a filter element change is required. An electric warning system can be incorporated to set off an alarm to indicate a clogged filter.

All filter cartridges should be sufficiently strong to prevent collapse or rupture under the most adverse operating conditions.

Some manufacturers provide an external filtering circuit and do not require a suction filter. A 10 micron filter with by-pass and warning device are recommended.

Heat Exchanger: (Figure 11-3, Item 10)

Provisions should be made in the system to insure that the maximum continuous operation temperature shall not exceed 180° F at the motor case drain. This may require the use of a heat exchanger in the reservoir return circuit, dependent upon the specific duty cycle and design of the machine. Generally, the optimum heat exchanger size should be capable of dissipating 20-25% of the maximum transmission input horsepower.

The fluid restriction resulting from the case drain lines and heat exchanger should not exceed manufacturers recommendations at normal system temperature. This may require the use of a pressure by-pass around the heat exchanger (Figure 11-4, Item 14).

Hydraulic Fluid

The hydraulic fluid used in the system should be selected using the guidelines given in the fluid recommendations section.

Hydraulic Lines: (Figure 11-3, Items 6,7,8,9,11, & 13)

The hydraulic lines selected should be compatible with good hydraulic practices regarding length, diameter, pressure capabilities, bend radii, fluid compatibility and transmission operating limits (see Figure 11-4).

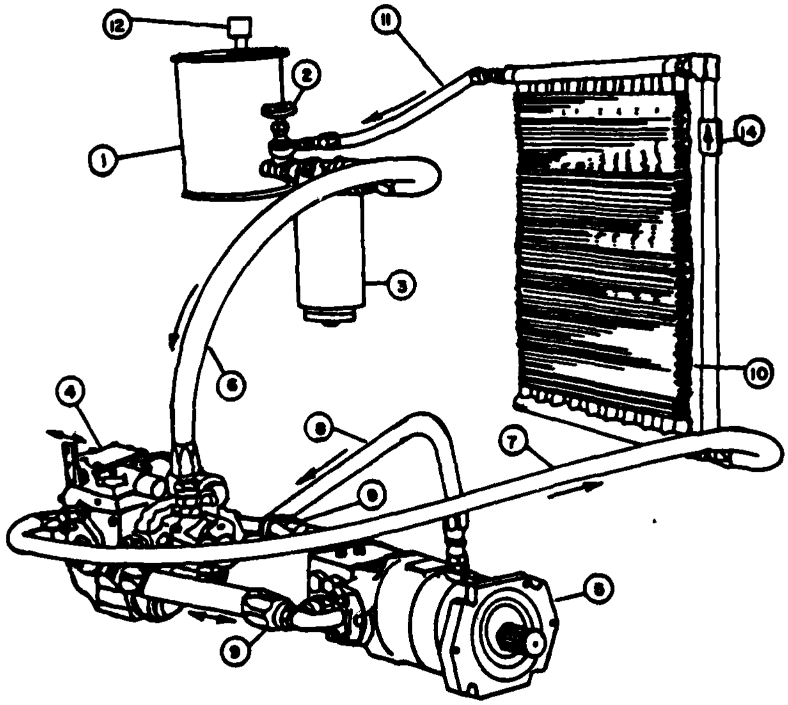
4.0 HYDRAULIC FLUIDS (Petroleum and Fire-Resistant Types)

In discussing fluids for hydraulic systems other than aerospace (which speaks a language all its own) we have to think in two general areas: petroleum fluids and fire-resistant fluids. These, in turn, break down into a number of types with different properties.

The reason we stick to the high sounding name of "petroleum fluid" rather than just plain "oil" is to emphasize that it is a special formulation with the additives to make it suitable as a hydraulic fluid. Primarily these additives inhibit or prevent rust, oxidation, foam, and wear.

It is hard to beat a high grade, inhibited petroleum fluid if you are not concerned with fire hazards. It has been the workhorse of hydraulic fluids for years. But there are new developments in petroleum fluids that will be discussed later.

Fire-resistant fluids, as the name implies, overcome the hazard of fire caused by a broken hydraulic line spraying



1 RESERVOIR
2 SHUT OFF VALVE
3 FILTER
4 VARIABLE DISPLACEMENT PUMP
5 FIXED DISPLACEMENT MOTOR
6 INLET LINE
7 PUMP CASE DRAIN LINE
8 MOTOR CASE DRAIN LINE
9 HIGH PRESSURE LINES
10 HEAT EXCHANGER
11 RESERVOIR RETURN LINE
12 RESERVOIR FILL CAP OR BREATHER
14 HEAT EXCHANGER BY PASS VALVE

PLUMBING INSTALLATION
VARIABLE PUMP — FIXED MOTOR
FIGURE 11-3

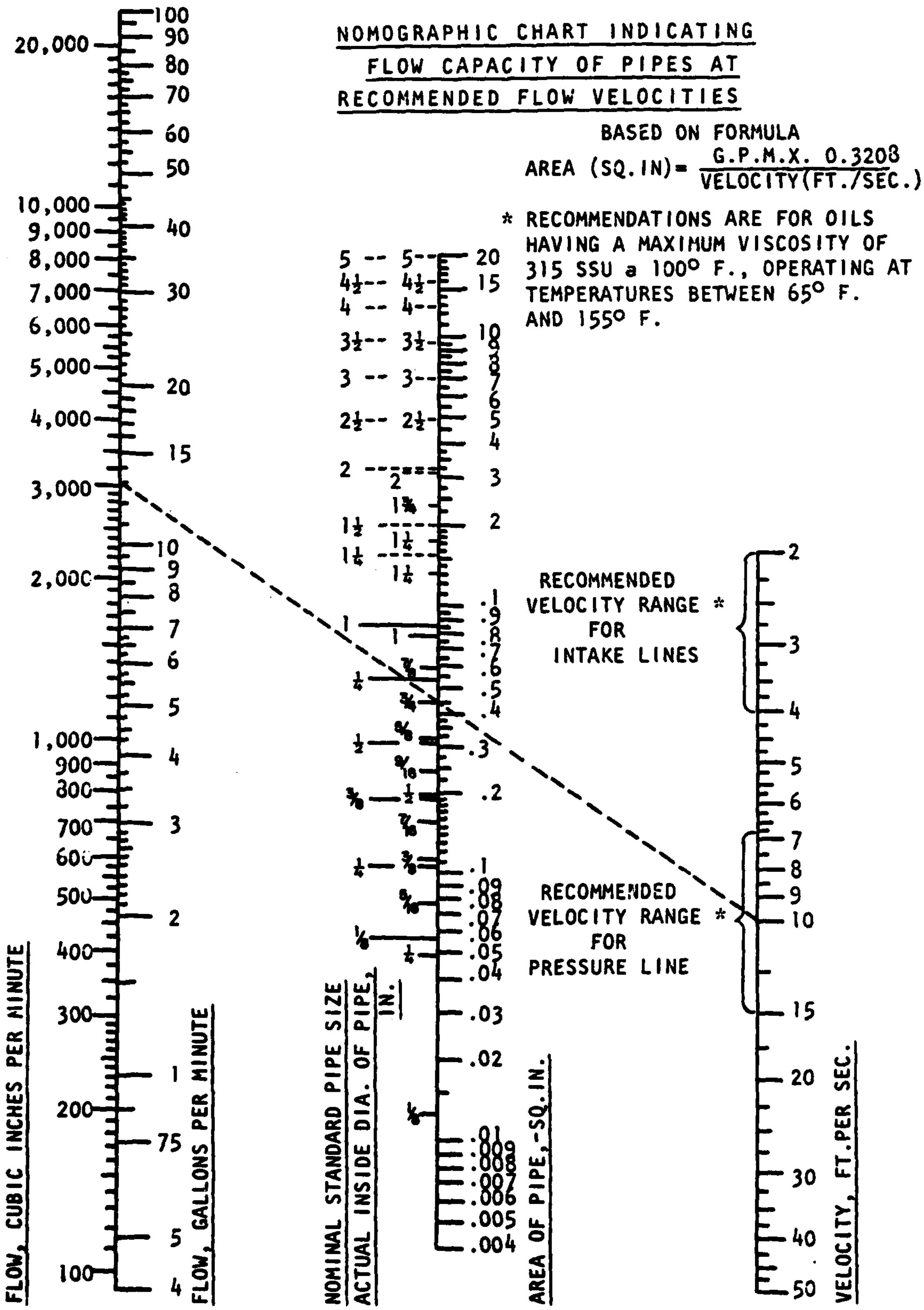

NOMOGRAPHIC CHART INDICATING
FLOW CAPACITY OF PIPES AT
RECOMMENDED FLOW VELOCITIES
BASED ON FORMULA
AREA (SQ.IN) = G.P.M.X. 0.3208 / VELOCITY (FT./SEC.)
* RECOMMENDATIONS ARE FOR OILS HAVING A MAXIMUM VISCOSITY OF 315 SSU a 100° F., OPERATING AT TEMPERATURES BETWEEN 65° F. AND 155° F.
RECOMMENDED VELOCITY RANGE * FOR INTAKE LINES
RECOMMENDED VELOCITY RANGE * FOR PRESSURE LINE
FLOW, CUBIC INCHES PER MINUTE
FLOW, GALLONS PER MINUTE
NOMINAL STANDARD PIPE SIZE
ACTUAL INSIDE DIA. OF PIPE, IN.
AREA OF PIPE,-SQ.IN.
VELOCITY, FT.PER SEC.

FIGURE 11-4

petroleum fluid into a pot of molten metal, hot manifold, heat-treating furnace or other ignition source.

Three types of fire-resistant fluids are in general use-straight synthetic, water-glycol, and water-in-oil emulsion. Properties of viscosity, viscosity index, lubricity, and fire-resistance vary. Fire-resistance is usually achieved at the cost of other properties relative to petroleum fluids, but if you need the insurance, design around these characteristics to maintain top efficiency.

4.1 Fluids Vocabulary

a. Lubricity--The fluid must keep friction low, maintain an adequate film between moving parts to prevent wear of pump, bearings, vanes, gears, pistons, and rods. Increasing pressures and consequently closer tolerances makes lubricity even more important.

b. Viscosity--Fluid "thickness" or resistance to flow. Pump manufacturers specify this according to clearances, speeds, temperatures, and suction characteristics. The fluid must be thin enough to flow freely, heavy enough to prevent wear and leakage.

Viscosity might not be so critical in selecting a hydraulic fluid except that it varies with temperature. Fluid thickens when it cools; thins out as it heats up. Because some hydraulic systems must work under wide temperature extremes, viscosity range is important. Generally pump manufacturers hate to see you try to start their pumps when viscosity is higher than 4000 SUS (Saybolt Universal Seconds) (870 cs.), and they usually want to keep above 70 SUS (13 cs.), during operation.

c. Viscosity Index--This measures the rate of viscosity change with temperature: the higher the index, the more stable the viscosity as temperature varies.

Viscosity index can be improved by additives, usually polymers, but because these polymers have molecules consisting of a long chain of atoms, they are susceptible to shearing and loss of effectiveness. So the choice of a VI improver is a compromise between its ability to raise VI and its resistance to shearing.

d. Rust resistance--Moisture gets into petroluem fluids by condensation and by contamination of the reservoir. Rust inhibitors and preventitives combat the effects of moisture. Obviously they are very important in water-in-oil emulsions and water-glycol fluids.

e. Oxidation resistance--Air, heat, and contamination all promote fluid oxidation which forms sludges and acids. Oxidation inhibitors delay the process.

f. Foaming resistance--Although control of foaming depends largely on reservoir design, antifoaming additives in the fluid help too.

4.2 Petroleum Fluids

One of the most dramatic developments in hydraulic fluids is the emergence of "new generation" petroleum fluids. These are an outgrowth of the use of motor oils in hydraulic systems of mobile equipment. Because they were readily available, motor oils have long been used in hydraulic systems of construction equipment, farm tractors, and other engine-driven vehicles. One very important-benefit that showed up was increased pump life: additives in these motor oils reduced wear. As pump manufacturers recognized this and ran their own tests, they actually recommended the motor oils and began to work with them in industrial applications.

The anti-wear characteristics of these new generation fluids are especially important as pressures move up and these fluids have already solved some tough problems in pumps operating overloaded at high temperature or with contaminated fluid.

These motor oils have been referred to as "crankcase oils," "MS motor oils" (because they meet the MS test sequences published in 1959 and revised in 1961 by the American Society for Testing and Materials), and "anti-wear hydraulic fluids." Naturally the fluid suppliers are putting their own brand names on their individual products so you can also identify them that way.

4.3 Fire-Resistant Fluids

With all the considerations of system performance and life, still another one overshadows all the rest in selecting a fluid: fire hazard. Engineers in a number of fields now realize that the finest hydraulic fluid in the world should not be used if a line break could cause a disastrous fire.

4.4 Synthetic Fluids

Generally considered the ultimate in fire-resistant fluids the phosphate esters, phosphate ester base, and chlorinated hydrocarbon base fluids are outstanding for their ability to lubricate at high pressures, and they are particularly recommended where pumps have heavily loaded antifriction bearings. But, they still have room for improvement, notably in viscosity index. VI improves help, but these degrade with shearing action in the pumps.

Although straight synthetics work better than water containing fluids in systems which run hot, the medium viscosity fluids are not generaly recommended for very low temperature systems without auxiliary heating. Start-up temperature should

be at least 40° F. Most suppliers have low viscosity fliuds for low temperature applications. Thus, the lower viscosity index does not have to be a problem if your temperature does not vary widely or if you control it.

A big factor in selecting straight synthetic fluids is that they are not compatible with seals used for petroleum fluids. However, seals of butyl rubber, Viton, EP (ethylene-propylene) rubber, silicone, Teflon, and nylon are suitable and are available. Suppliers of straight synthetic fluids suggest changing only the dynamic seals at the time of conversion. Static seals may then be changed during routine maintenance.

4.5 Water-Glycol Fluids

Low temperature characteristics are very good, especially where fluid is stored at low temperature. In operation, temperature must be limited to avoid abnormal loss of water which should make up about 35% to 50% of the solution.

With water-glycol fluids (and emulsions, too) you have to keep water content up. Fluid suppliers have published instructions for this and will make their laboratory facilities available for special cases. Since viscosity of water-glycol fluid increases with water loss, you can check water content by checking viscosity and this is recommended at 3-to 6-month periods. Viscosity can be measured with a viscosimeter. A table tells you how much water (distilled or de-ionized) to add to bring viscosity back to normal.

Because alkalinity for vapor phase inhibitor is also reduced with loss of water and with contamination, this also has to be checked and adjusted. Alkaline reserve is determined by a lab test using easily available equipment and materials. To bring up alkaline content, you add morpholine or other chemicals available from the fluid supplier.

Fluid suppliers preach that you will have to make adjustments in your maintenance practices. General shop organization and cleanliness is the first step.

4.6 ASTM Viscosity Classification for Industrial Systems

In recent years there has been considerable interest on the part of users of industrial lubricants for a viscosity classification system which would be adaptable to IBM machines for statistical listing. This is especially needed by large companies who may purchase hundreds of lubricants made by scores of manufacturers.

A system that has been adopted by ASTM is shown in Table 1. The purpose of this standard is two-fold: (1) To establish a series of standard viscosity levels so that lubricant suppliers, lubricant users, and equipment designers will have a

uniform and common basis for designating, specifying or selecting the viscosity of industrial fluid lubricants; and (2) To eliminate unjustified intermediate viscosities, thereby reducing the total number of viscosity grades used in the lubrication of industrial equipment.

This standard provides a suitable number of viscosity grades, a uniform reference temperature and a uniform viscosity tolerance and nomenclature system for identifying the viscosity characteristics of each grade.

The new standard implies no evaluation of lubricant quality and applies to no property of a fluid other than its viscosity at the reference temperature. It does not apply to those lubricants used primarily with automotive equipment and identified with an SAE number.

Complete details on this system have been published "For Information Only" in Appendix XXIII, pages 833-35, Volume 17, of the ASTM 1964 Book of Standards.

TABLE 1

ASTM VISCOSITY STANDARD FOR
INDUSTRIAL FLUID LUBRICANTS

ASTM Viscosity Grade Number	Saybolt Viscosity @ 100°F Maximum	Minimum
7000	7700	6330
4650	5115	4185
3150	3465	2835
2150	2365	1935
1500	1650	1350
1000	1100	900
700	770	630
465	512	419
315	346.5	284
215	237	194
150	165	135
105	115.5	94.5
75	82.5	61.5
60	66	54
40	44	36
32	35.2	28.8

(A) ASTM viscosity grade numbers are nominal viscosities at 100° F, expressed in Saybolt Universal Seconds (SUS).

(B) Viscosity range of each grade to be ± 10% the above nominal viscosity values.

(C) This standard implies no evaluation of quality.

5.0 CLEANLINESS OF HYDRAULIC SYSTEMS

In order to obtain the longest possible service from a hydraulic oil, it is essential that the oil be kept clean and free from contaminants. This applies not only to the time that the oil is in actual service, but also during storage and installation.

5.1 Storage and Handling

Refiners and marketers of hydraulic oils are particularly careful to assure that the oil is absolutely clean when it is delivered to the customer. The customer should likewise exercise equal care to assure that the oil is just as clean when it is installed in the hydraulic system. Contamination of the oil by such materials as dust, water and lint can easily be prevented by observing a few simple precautions:

Store drums on their sides--indoors if possible, but in any event under a shelter of some sort.

Before opening a drum, clean the top so that no dirt can fall in the oil.

Any containers or hoses used in transferring the oil from the drum to the equipment reservoir should be thoroughly clean.

The oil should be filtered as it enters the reservoir. If the fill pipe on the reservoir does not include a filter screen, a funnel equipped with a 200 mesh screen will be satisfactory.

These rules are simply common sense and should be second nature to anyone handling hydraulic oils; yet the amount of trouble that develops because they are not observed is surprisingly large.

5.2 Fluids in Service

Hydraulic system manufacturers exert every effort to design their systems to minimzie the possibility of contamination. However, despite all precautions, contamination will occur and the extent will vary with the type of system, its condition and the nature of its application. Greases can gain an entry through packing glands or piston rods. Atmospheric dust can enter through the reservoir breather. Surface paints, rustproof preparations, gasket cements and pipe sealing compounds may be suspended in the oil.

Contaminants can be classified broadly as follows:

1. Extraneous materials such as dust, dirt, rust, scale, etc., which are in no way attributable to the oil.

2. Soluble or insoluble products of oil deterioration resulting from oxidation or polymerization reactions.

Although contaminants of either classificaiton can adversely affect the operation of a hydraulc system and therefore should be removed, the oil itself, in the case of contamination by extraneous material, will probably be suitable for continued service. Consequently it is simply a matter of economics as to whether it is cheaper to discard the contaminated oil or remove the contaminants from the oil so it may be re-used. When contamination due to oil deterioration is too great the hydraulic oil should be replaced.

There are various basic methods, both continuous and batch, that can be employed to remove contaminants and purify oil. The equipment user must decide which is most practical and economical in each case. With a batch method, the oil must be removed from the hydraulic system for treatment. This necessitates the use of another charge of oil if the equipment is to operate during the interim. With continuous methods, the purification or reconditioning equipment is an integral part of the hydraulic system and the oil does not have to be removed from service. Another disadvantage of a batch method is that the contaminants are allowed to circulate and build up in the hydraulic system until the oil is removed for treatment.

The principles of some of the methods that can be employed to recondition hydraulic oils are described as follows:

5.3 <u>Gravity Settling</u>

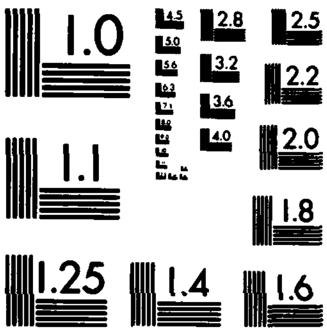
A relatively simple means of removing a major portion of any material suspended in a hydraulic oil involves transferring the oil to a setttlig tank and permitting it to remain undisturbed for a period of time during which the insoluble contaminants will separate by gravity. Any water present will also settle out unless it has formed a stable emulsion with the oil.

Horizontal tanks with V-shaped or sloping bottoms serve as the most effective settling tanks. Although separation of contaminants will occur at room temperature, the process can be hastened by heating the oil and maintaining it at a temperature of 120-160^{o}F. To accomplish this, the settling tank should have either jacketed walls or hot water coils. Care should be taken to prevent the oil from being overheated or oxidation might take place. It will be observed that practically no settling will occur while the oil is being heated due to the con-

AD-A132 513 HANDBOOK OF OCEANOGRAPHIC WINCH WIRE AND CABLE TECHNOLOGY(U) RHODE ISLANDUUNIV KINGSTON GRADUATE SCHOOL OF OCEANOGRAPHY A H DRISCOLL 18 OCT 82 5/5

UNCLASSIFIED N00014-82-G-0116 F/G 8/10 NL

END



MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

vection currents created. Settling times may vary but it is recommended that the oil be allowed to remain in the tank for at least ten days.

5.4 Centrifuging:

Centrifuging will achieve the same results as gravity settling but is much faster, since the separating force is several thousand times that of gravity. As with settling, centrifuging will remove foreign impurities, insoluble oxidized material, dirt and water, but will not separate liquids that are mutually soluble in the oil, or impurities of the same gravity as the oil.

Water may be added to the oil to be centrifuged. This will aid in the removal of impurities which have gravities of the same order as that of the oil and also any soluble acidic materials which may be more soluble in water than in oil. Care should be observed in applying wet centrifuging to inhibited oils as certain types of additives may concentrate at the oil-water interface and a portion of them will be removed. Dry centrifuging will not remove additives.

5.5 Continuous Reconditioning

Purification equipment integrated with the hydraulic system for continuous reconditioning of the oil provides filtration for the removal of insoluble and some soluble contaminants, and heat for vaporizing moisture. The location of filters in any hydraulic system depends on many factors and the manufacturer should be consulted as to the most satisfactory arrangement. It is customary for the filters to be located at some point in the system after the pump. In some instances the filterig equipment may be so installed that the full pump discharge passes through the filter. In other installations only a portion of the oil passes through the filter, the balance being by-passed. This arrangement is becoming increasingly popular because, should the filter become plugged, operation of the hydraulic system is not impaired and there is no danger of the filter being back-washed. In some instances a separate pump may be installed to circulate oil from the reservoir, through the filter and back to the reservoir.

Three types of filters are available, characterized by the nature of the filtering medium employed. When the elements become partially clogged with contaminants, they can either be cleaned or replaced.

Metallic or mechanical filters contain closely woven metal or discs of metal as the filtering elements. They will remove coarse, solid contaminants but not soluble oxidized material, water, or finely divided contaminants such as dust and some insoluble oxidation products. These filters can be used

safely with inhibited oils, as they will not remove additives.

Absorbent (inactive) filters contain materials such as cotton waste, wood pulp, wool yarn, felt, flannel, cloth, paper, mineral wool, quartz, diatomaceous earth and asbestos as filtering elements. Not only do these materials remove coarse contaminants but they will also take out fine particles, water, and water soluble impurities. They will not remove soluble oxidation products nor additives from inhibited oils.

Absorbent (active) type filters remove impurities and contaminants by chemical attraction in addition to purely mechanical means. Boneblack, charcoal, chemicaly-treated paper, Fuller's earth and other active-type clays are examples of materials used as filtering elements. In addition to both coarse and fine insoluble particles, these filters can remove practically all insoluble sludge, water and soluble oxidized materials. Furthermore, they may remove cost additives used in inhibited hydraulic oils, and consequently considerable care should be exercised in their application.

5.6 Filtration for Fluid Power Systems

Properly designed filter systems plus systematic filter maintenance can eliminate at least 85% of the potential causes of hydraulic system failure. In addition, new filtration techniques to the 1-micron level greatly extend the life of system components.

Contaminant Sources: Most hydraulic systems recirculate the same oil. Although the systems are "closed" they are not dirt proof. Harmful dirt and foreign particles may be built in, introduced, or produced by wear. Built-in contaminants (or dirt) result largely from the manufacture of the equipment, and include core sand, weld spatter, metal chips, lint and abrasive dust. Introduced contaminants enter the system through seals, fluid-filler tubes, and breather caps in reservoirs. Lint and other foreign matter may enter when the system is opened. Fluid used for replenishing might contain dirt. Wear contaminants include small particles of metal and sealing materials which result from wear of moving parts within the system. Fluid breakdown can result in the formation of sludge and acids. It results from chemical reactions within the fluid caused by water, air, heat, and pressure, as well as incompatible fluids. Sludge is not generaly abrasive; however, it is recognized as the source of resinous and gummy coatings on moving parts and can clog passages. Acids can pit and corrode critical moving parts.

Contaminant Effects: Each dirt particle in the system is the "abrasive seed" that produced additional dirt particles as it passes through pumps, valves, and actuators. Clearances in hydraulic components may be extmely close, as small as 0.0002-0.0005 inches (5-12 microns) or less. When abrasive particles of dirt enter the space between moving parts they

score or hone the surfaces to greater clearances, and coincidentally produce additional particles. Thus the process continues at ever increasing speed as the system runs.

Common Results of Contamination Are: 1. Internal leakage (or slippage) which lowers the efficiency of pumps, motors and cylinders and decreases the ability of valves to control flow and pressure accurately. It also wastes horsepower and generates heat. 2. Sticking of parts due to sludge or silting (collection of fine particles). 3. Premature failure of components.

5.7 Filter Selection Factors

Degree of Filtration: Any hydraulic system should have some degree of filtration. Systems in daily use should be filtered to the 1-micron level if possible to minimize maintenance and greatly extend component life. The savings from decreased maintenance and from perhaps several years additioal component life make the job of justifying the expense of fine filtration (1 micron nominal) an easy one.

In systems which are back up or only irregularly used, perhaps the filtration can be relaxed and used only to prevent the possibility of catastrophic failure and the effects of micronic system abrasion and wear ignored. In this case a 25-micron level could be established as minimum.

In selecting filters, the following conditions should be carefully understood:

- Pressure Drop. All system components through which there is flow have a pressure drop. This drop is the net pressure required for the fluid to flow from the inlet to the outlet of the component. In filters, this includes the pressure drop across the housing and the filter element. It varies with flow rate, fluid viscosity, and density. In making a filter selection, the maximum allowable pressure drop when the filter element is dirty, the operation temperature, and the lowest temperature of use, must be considered.

- Dirt Capacity. This is the maximum amount of contaminant that can be collected by a barrier-type filter element without producing a pressure drop which affects hydraulic system function or damages the filter element. Initially, pressure drop increases only slightly with increasing contaminant collection. However, as the contaminant collected by the filter increases, the quantity and size of available flow passages decrease, so that more of the smaller particles are stopped. The filter eventually reaches a point beyond which additional contaminant collection causes a rapid increase in pressure drop. The abruptness of this rise depends

on the type and fineness of the media used in the filter elements. Filter-element servicing recommendations can be based on either a time interval or pressure-drop reading. Because many factors which cannot be accurately predicted affect rate of contaminant collection, service-interval recommendations based upon hours of operation should be conservative. Alternately, for full flow filters differential pressure readings indicate the actual amount of contaminant collection, and are thus a more realistic basis for element servicing.

System Pressure. Filter housings must be capable of withstanding maximum system pressure when installed in the pressure line. The housing, elements, and bypass valves must withstand fatigue from cycling and pressure surges.

Temperature. In filter selection, start-up and normal operating temperatures, as well as maximum temperature that may be encountered, should all be considered as they apply to housings, seals, bypass valve characteristics and filter elements.

Other Factors. The environment in which the equipment operates is important, particularly if the reservoir is vented to atmosphere. The degree of filtration provided by the breather-cap fliter and filler cap should be considered in order to prevent as much dust and other contaminants as possible from entering the reservoir.

With increased emphasis today on prevention of machine downtime, the need for much higher standards of fluid cleanliness is evolving. Since pumps and valves have clearances on the order of 5 microns (0.0002 inches) it is easy to understand the need for maintaining particle size below this size. To meet this demand, filters now are available capable of filtering down to the 1-micron level and up to more than 100 GPM and 5,000 psi pressure flows.

Coincidental with the demand for cleaner hydraulic fluid has been the increased use of low pressure portable recirculating units capable of reducing the contaminant level of the oil to the 1-micron level or SAE Class 0 described in Table 1.

Placement of filters within a hydraulic system requires a good understanding of all factors. A filter at the pump inlet cleans all fluid before it encounters any moving part, however a filter at this location may cause excessive pressure drop and pump cavitation. Sometimes a coarse filter is placed at the pump inlet merely to protect the system against bulky objects which may get into the reservoir, which could cause a catastrophic failure of the pump.

A filter immediately downstream from the pump protects all components except the pump itself, and removes all debris created by the pump. As was pointed out above, the current practice of the use of high performance filters with 1-micron nominal capability is at this location.

A filter in the return line has the advantage of operation under minimum pressure, and if the oil is clean such a filter will maintain it in good condition. However, all particles will have passed through almost all components before they reach this filter.

Filter maintenance is as cruicial to the efficient operation of a hydraulic system as is the presence of the filter itself. The following check list is provided as a guide to the maintenance of the hydraulic filters in the system.

1. Set up a filter maintenance schedule and follow it diligently.

2. Inspect filter elements that have been removed from the system for signs of failure which may indicate the need for shortening the service interval and the possibility of other system problems.

3. Do not return to the system any fluid which has leaked out.

4. Always keep the supply of fresh fluid covered tightly.

5. Use clean containers, hoses, and funnels when filling the reservoir.

6. Use common sense precautions to prevent entry of dirt into components that have been temporarily removed from the circuit.

7. Make sure that all clean-out holes, filler caps, and breather cap filters on the reservoir are properly fastened.

8. Do not run the system unless all normally provided filtration devices are in place.

9. Make certain that the fluid used in the system is of a type recommended by the manufacturers of the system of components.

6.0 HYDRAULIC SYSTEM MAINTENANCE

• <u>Fluid</u>: Laboratory analysis of a carefully obtained sample of oil is the best method of determining the condition of the fluid. Generally, a fluid change interval of 2000 hours is

adequate with a sealed reservoir system. A more frequent fluid change is required if the fluid has become contaminated by water or other foreign material or has been subjected to abnormal operating conditions, such as overheating.

An open reservoir system with an air breathing filler cap might require the fluid to be changed every 1000 hours.

- Filter: As a general recommendation, with a sealed reservoir system, the 10 micron filter should be changed yearly or every 1,500 hours, whichever occurs first. With an open reservoir system utilizing an air breathing filler cap, the filter should be changed every 1000 hours. An alarm system on the filter will ensure proper filter change intervals.

- Reservoir: The reservoir should be checked weekly for the proper fluid level and the presence of water in the fluid. If fluid must be added to the reservoir, use only filtered or strained fluid. Drain any water as required. A heater system in the reservoir can be installed to preheat the fluid in cold weather and help evaporate moisture.

- Hydraulic Lines and Fittings: Visually check daily for any fluid leakage. Then, repair or replace as required.

- Heat Exchanger: The heat exchanger core and cooling fins should be kept clean at all times for maximum cooling and system efficiency. Inspect weekly for any external blockage and clean as required.

6.1 Drain Schedules

By using the reconditioning methods just described, the service life of an oil can be extended appreciably. However, it must be understood that the treatments do not transform a used oil into a new one; they are simply means of assuring that the longest life possible can be realized from a given oil. Eventually, after prolonged use, the condition and nature of the oil will have changed to such an extent that it is no longer suitable for safe and efficient operation. When this stage is reached, it is essential that the oil be replaced with a new charge. Before installation, the entire system should be examined and, if necessary, cleaned in accordance with procedures to be described.

The frequency with which used oil should be drained and replaced depends on the nature of the oil and the operating conditions to which it is subjected. Consequently it is impossible to establish a drain schedule that will apply categorically to all systems. Visual inspection of the oil to note any change in appearance, such as darkening or thickening, may serve as a rough guide to indicate that the need for a change is imminent. However, periodic testing of the oil is the safest and best way to determine when its condition is such that it should

be replaced. The oil supplier should be consulted with reference to establishing the intervals at which the oil should be tested, as the frequency of testing will not be a fixed period but will increase as the condition of the oil starts to change appreciably. If laboratory facilities are not available, the oil supplier will usually arrange to have the oil tested at his own laboratory. The oil should be changed when characteristics such as viscosity and acidic properties begin to increase at an accelerated rate. If the oil is drained at the proper time, the hydraulic system should be in such condition that subsequent cleaning will be relatively easy. On the other hand, if the oil is allowed to continue in service after it should have been replaced, cleaning and flushing of the system may be time consuming and laborious.

6.2 Cleaning and Flushing

Satisfactory performance of hydraulic systems depends on the use of clean oil in clean equipment. The importance of having the systems absolutely clean before introducing a charge of oil cannot be emphasized too strongly.

The cleaning of hydraulic systems cannot be reduced to a simple routine practice since the procedures used must be varied to suit the conditions encountered in each individual case. Some of the methods normally employed are described as follows:

a) New Systems

In new equipment, the hydraulic system is often an integrated unit and is completely sealed at the time of shipment. Manufacturers of such equipment take great care to see that no contaminants enter the system. In addition, they are invariably tested at the factory with a suitable oil containing a rust inhibitor, which ensures rust protection during shipment. Therefore, in such cases it is not usually necessary to flush out the system upon receipt, for the machine is ready for the initial charge of hydraulic oil. (To be sure, check this point with your machine supplier.)

Manufacturers who, because of the size of their equipment, must ship machines in parts always cap or flange off exposed ends of hydraulic pipes. These should not be removed until assembly is made and great care must be exercised to see that no dirt, etc., enters the exposed ends during assembly. Examination of such systems will show if they should be flushed out before initially charging the hydraulic oil.

Manufacturers who construct hydraulic systems exercise precautions to see that all mill scale, rust, and other foreign matter are removed from pipes and other parts. In addition, all core sands from castings, and bonding sand must be removed. This is accomplished normally by immersing the parts

in dilute sulfuric acid, a process called "pickling." After removal, the acid remaining is neturalized, often by dipping in a 5 per cent soda-ash solution. Next the parts must be washed in water and dried by blowing with steam or air. Sand blasting is also used to remove rust and scale. After fabrication (bending, welding, flanging, etc.) the interior of all piping is wire brushed to remove any materials introduced after cleaning, then blown out with air to remove the loosened material. After fabrication and cleaning, all interior surfaces are then rust-proofed in some manner, usually with a rust inhibited hydraulic oil or a petroleum type rustproofing compound. If the latter is used, this material must be removed before the intitial charge of hydraulic oil is made. Since the compounds used are normally oil soluble, such compounds can be removed by flushing, as outlined below. In any case, the manufacturer should be consulted to determine the best method of removing these compounds. After rustproofing, all openings are capped or flanged so that no dirt will enter.

b) Cleaning Hydraulic Systems After Use

If deposits tend to accumulate in a hydraulic system, they are chiefly oil oxidation products which gradually become insoluble in the oil, together with contaminants, such as dust or lint, mineral matter from the metal parts of the system and condensation which binds the insoluble material into sludge-like emulsion. The nature and amount of deposit present in a particular system may vary widely. Inspection may show any condition between a viscous oil film and a hard solid deposit which completely chokes oil passages.

Deposits in a hydraulic system are of two general types. If they are oily in nature and present as films or light emulsion coating on metal surfaces, the system can be cleaned by flushing with a suitable flushing oil. In systems containing appreciable amounts of solid or semi-solid deposits (a condition which rarely, if ever, exists with modern inhibited hydraulic oils), flushing will not remove the accumulation under ordinary circumstances. In such cases the only alternative is to dismantle the system and clean it manually.

c) Lightly Sludged Systems

Deposits, oily in nature and present as film or light emulsion type coatings on metal surfaces, can normally be removed by flushing.

A suitable flushing oil should have the following properties:

a. A viscosity sufficiently high to adequately lubricate moving parts and to ensure continued suspension of particles during circulation, yet low enough to have high solvent power. Usually an oil with a viscosity

within the range of 70 to 110 SUS at 100°F is considered satisfacatory.

Suitable solvent power to remove oily material (in addition to petrolatum type rustproof compounds, etc., from new sytems). Generally naphthene base oils are more satisfacatory than paraffin base oils because of their higher solvency. It is not felt that so-called "gum or varnish solvents" are necessary in flushing oils.

c. Rust inhibitors which coat all metal surfaces with a film capable of preventing rust formation.

d. Oxidation inhibitors which permit any flushing oil trapped in the systems to blend with the final charge of inhibited hydraulic oil without adversely affecting the latter's potential service life.

The advice of the oil supplier should again be sought regarding the selection of a suitable flushing oil. Usually he will be able to furnish a product which has been prepared especially for this purpose and which will provide adequate cleaning efficiency combined with the other necessary characteristics.

Most solvents and chemical cleaners on the market today are not recommended for use in hydraulic systems for several good reasons. Among these is the fact that some do not offer sufficient lubricating value with the result that moving parts, and particulary the pump, are damaged. A second reason is that it is very difficult to remove all traces of the solvent or cleaner from the system. Just a trace of some of the commercial chlorinated solvents may be sufficient to reduce the oxidation resistance of premium grade inhibited hydraulic oils to the level of that of a straight mineral oil. Also, in the presence of a small amount of water, some of these solvents will become very corrosive to steel and copper.

Following is a suggested procedure for flushing hydraulic systems:

Drain the hydraulic oil from the system and clean the filters and strainers. It is also advisable to remove as much sludge as possible from the reservoir.

Charge the system with the recommended flushing oil and operate the equipment. During the circulation a portion of the oil should be by-passed continually through a filter and the rate of deposit removal closely observed. It is important that the valves be so manipulated that the flushing oil goes through all lines. If possible, a flushing pump should be used to produce turbulent flow to assist in the cleaning process. The minimum flow of the flushing pump should exceed the maximum flow of the permanent system.

The time necessary to clean the system will vary, depending on the condition of the equipment. Usually from 4 to 48 hours is sufficient for most systems.

After the flushing oil has been drained from the system, the equipment should be inspected to see that it is in satisfactory condition to receive a new charge of hydraulic oil. If deposits still persist, the system will have to be refreshed or dismantled and cleaned manually.

If the flushing oil can be drained completely from the equipment, the system can then be charged with the new hydraulic oil. If appreciable quantities of flushing oil remain after draining, this should be flushed out with the hydraulic oil to be employed. This hydraulic oil can then be retained for similar application on other systems or used as a general machine lubricant after it has been suitably filtered and reclaimed.

d) Heavily Sludged Systems

When appreciable amounts of solid or semi-solid deposits are present in a hydraulic system, flushing will not remove the accumulation under ordinary circumstances. Such systems should be dismantled and cleaned mechanically.

In the mechanical removal of semi-solid or solid deposits, it is generally necessary to dismantle the system because of the inaccessibility of many parts. The method of removal depends largely on the size of the lines and other equipment, as well as the type of deposits. Scraping, wire brushing and even rotary boiler tube cleaners have been resorted to. The success of such methods will be in direct proportion to the precautions taken in seeing that all parts are suitably cleaned. As a result, it is best to have someone in charge of such operations who is experienced in this line of work.

As each part of the system is mechanically cleaned it should be blown out with air; then the freshly cleaned metal surface should be coated with a hydraulic oil containing a rust inhibitor. Normally the same oil finally used in the system should be used for this purpose, providing the oil is rust inhibited. Freshly cleaned metal surfaces have a tendency to rust quickly, and as a result it is imperative that they be rust protected. After all parts have been cleaned the system should be re-assembled, using care to see that no dirt, lint, pipe thread compound, etc., gets into the system.

As a final step it is advisable to flush out the system, using a flushing oil, as previously described.

7.0 TROUBLE SHOOTING HYDRAULIC SYSTEMS

Hydraulic mechanisms are precision units and their continued smooth operation depends on proper care. Therefore, do not neglect hydraulic systems. Keep them clean. Change the oil and oil filter (if present) at established intervals.

If, in spite of these precautions, improper operation does occur, the cause can generally be traced to one of the following:

1. Use of the wrong viscosity or type of oil.
2. Insufficient fluid in the system.
3. Presence of air in the system.
4. Mechanical damage or structural failure.
5. Internal or external leakage.
6. Dirt, decomposed packing, water, sludge, rust, etc., in the system.
7. Improper adjustment.
8. Oil cooler plugged, dirty, or leaking.

Some possible causes of specific troubles which may be encountered and their remedy are indicated in the following pages.

7.1 Improper Operation of Pumps

A. Failure of pump to deliver

Possible Causes	Remedy
1. Low fluid level in reservoir.	1. Add recommended oil and check level on both sides of tank baffle to be certain pump suction line is submerged.
2. Oil intake pipe or suction filter plugged.	2. Clean filter or otherwise remove obstruction.
3. Air leak in suction line, preventing priming or causing noise and irregular action of control circuit.	3. Repair leaks.
4. Pump shaft turning too slowly to prime itself (vane type pumps only.)	4. Check minimum speed recommendations in manufacturers' descriptive literature.

5. Oil viscosity too heavy to pick up prime.	5. Use lighter viscosity oil. Follow manufacaturers' recommendations for given temperature and service.
6. Wrong direction of shaft rotation.	6. Must be reversed immediately to prevent seizure and breakage of parts due to lack of oil.
7. Broken pump shaft or parts broken inside pump. Shear pin or shear linkage broken.	7. Refer to manufacaturers' literature for replacement instructions.
8. Dirt in pump.	8. Dismantle and clean.
9. On variable delivery pumps the stroke is not right.	9. Check pump manufacaturers' instructions.

B. No pressure in the system

Possible Causes	Remedy
1. Pump not delivering oil for any of the above reasons.	1. Follow remedies given above.
2. Relief valve not functioning properly. (a) Valve setting not high enough. (b) Valve leaking. (c) Spring in relief valve broken.	2. See below (a) Increase pressure setting of valves. (b) Check seat for score mark and reseat. (c) Replace spring and readjust valve.
3. Vane or vanes stuck in rotor slots (vane type pumps only).	3. Inspect for wedged chips or sticky oil.
4. Head too loose (very infrequent).	4. Must not be tightened too tightly. See manufacturers' instructions before tightening.
5. Free re-circulation of oil to tank being allowed through system.	5. Directional valve may be in open-center neutral, or other return line open unintentionally.
6. Internal leakage in control valves or cylinder.	6. To determine location progressively, block off various parts of circuit. When trouble is located, repair.

C. Pump making noise

Possible Causes	Remedy
1. Partially clogged intake line, intake filter or restricted intake pipe.	1. Clean out intake, strainer or eliminate restriction. Be sure suction line is completely open.
2. Air leaks (a) At pump intake piping joints. (b) At pump shaft packing (if present). (c) Air drawn in through inlet pipe opening.	2. See below. (a) Test by pouring oil on joints while listening for change in sound of operation. Tighten as required. (b) Pour oil around shaft while listening for change in sound of operation. Follow manufacaturers' recommendations when changing packing. (c) Check to be certain suction and return lines are well below oil level in reservoir. Add oil to reservoir if necessary.
3. Air bubbles in intake oil.	3. Use hydraulic oil containing a foam depressant.
4. Reservoir air vent plugged.	4. Air must be allowed to breath in the reservoir. Clean or replace breather.
5. Pump running too fast.	5. Check recommended maximum speeds from manufacturers' descriptive bulletins.
6. Too high oil viscosity.	6. Use lower viscosity oil. Follow manufacturers' recommendations for given temperature and service.
7. Filter too small.	7. Capacity may be adequate only when just cleaned, and should have added capacity.
8. Coupling misalignment.	8. Re-align.

9. Pump head too loose, or a faulty head gasket.	9. Test by pouring oil over head, replacing gasket or tighten head as is necessary.
10. Stuck pump vane (vane type pump).	10. Inspect for wedged chips or sticky oil, and reassemble.
11. Worn or broken parts.	11. Replace

D. External oil leakage around pump

Possible Causes	Remedy
1. Shaft packing worn.	1. Replace.
2. Head of oil on suction pipe connection.	2. Sometimes necessary, but will usually cause slight leakage. Keep all joints tight.
3. Damaged head packing.	3. Replace.

E. Excessive wear

Possible Causes	Remedy
1. Abrasive matter in the hydraulic oil being circulated through the pump.	1. Install adequate filter or replace oil more often.
2. Viscostiy of oil too low at working conditions.	2. Check pump manufacturers' recommendations or consult your lubrication engineer.
3. Sustained high pressure above maximum pump rating.	3. Check relief or regulator valve maximum setting.
4. Drive misalignment or tight belt drive.	4. Check and correct.
5. Air recirculation causing chatter in system.	5. Remove air from system.

F. Breakage of parts inside pump housing

Possible Causes	Remedy
1. Excessive pressure above maximum pump rating.	1. Check relief or regulator valve maximum setting.
2. Seizure due to lack of oil.	2. Check reservoir level, oil filter and possiblity of restriction in suction line more often.

3. Solid matter being wedged in pump.	3. Install filter on suction line.
4. Excessive tightening of head screws.	4. Follow pump manufacturers' recommendations.

7.2 Improper Operation of Actuating Mechanisms

A. System inoperative

Possible Causes	Remedy
1. Any of the reasons listed above.	1. Follow remedies given above.

B. Mechanisms creep when stopped in intermediate position

Possible Causes	Remedy
1. Internal leakage in actuating cylinders or operating valves.	1. Replace piston packing or replace cylinder if walls are scored. Replace or repair valve.
2. Poppet in selector valve not seating.	2. Clean unit to remove foreign matter, then check cam clearance.

C. *Times of operation longer than specified*

Possible Causes	Remedy
1. Air in system.	1. Bleed system.
2. Internal leak in actuating cylinder or selector valve.	2. See Remedy for B; 1 and 2 (page 88).
3. Worn pump.	3. Repair or replace.
4. If action is sluggish on starting up, but somewhat less sluggish after operating temperatures have increased, or if action slows down after warm up (depending on equipment and circuit design), it is probable that viscosity of the oil is too high.	4. Consult pump manufacaturers' recommendations, or your oil supplier for correct oil viscosity.
5. Low auxiliary control pressure.	5. Control lines may be too small, particularly if they are long.

D. External oil leakage

Possible Causes	Remedy
1. End caps.	1. Tighten if possible or replace gasket if necessary.
2. Packing gland.	2. Tighten, or replace packing if necessary.

E. Abnormal packing gland wear

Possible Causes	Remedy
1. Cylinder not securely fastened to frame, causing vibration.	1. Tighten. This should be checked periodically.
2. Misaligment of cylinder and piston rod extension.	2. Check and correct.
3. Side load on piston rod.	3. Revise construction to eliminate side loads.

7.3 Improper Operation of Accumulator

A. Pressure from accumulator drops suddenly when position of selector valve is changed

Possible Causes	Remedy
1. Internal or external leak in accumulator	1. Repair leak or replace accumulator valve core (if present).

B. When pump is running pressure is normal, but when pump is stopped no pressure is available

Possible Causes	Remedy
1. Leaking gas valve or leaking check valve in hydraulic line.	1. Replace valve.

C. Sluggish response from accumulator

Possible Causes	Remedy
1. Stoppage of oil screen in accumulator (if present).	1. Dismantle accumulator and clean screen.
2. Gas precharge not sufficient.	2. Precharge according to manufacturers' instructions, also check for gas leaks.

Be sure all internal pressure is released before repairs are made on accumulators

7.4 Excessive Heating of Oil in System

A. Heating caused by power unit (reservoir, pump, relief valve and coolers)

Possible Causes	Remedy
1. Relief valve set at a higher presure than necessary, excess oil dissipated through increased slippage in various parts, or through relief valve or through throttle valve.	1. Reset relief valve to slightly above maximum pressure required for work stroke. Check manufacturers' recommendations for maximum pressure settings.
2. Internal oil leakage due to wear.	2. Repair or replace pump.
3. Viscosity of oil too high.	3. Follow manufacturers' recommendations for correct viscosity grade to be used at various temperatures.
4. Pumps assembled after overhaul may be assembled too tightly. This reduces clearances and increases rubbing friction.	4. Follow manufacaturers' instructions when reassembling.
5. Leaking check valves or relief valves in pump.	5. Repair.
6. Improper functioning of oil cooler or coolant is cut off.	6. Inspect cooler and see that it is working properly.
7. Automatic unloading control inoperative.	7. Repair valve.

B. Heating because of conditions in system

Possible Causes	Remedy
1. Restricted lines.	1. If lines are crimped, replace; if partially plugged for any reason, remove obstruction.
2. Large pump deliveries not unloaded properly.	2. Make certain that open-center valves are neutralized, and that any pressure-relieving valves are in the correct

	position. Only small pump volumes should be allowed to remain at high pressures when clamping or running idle for long periods of time.
3. Insufficient radiation.	3. Use artificial cooling.
4. Internal leaks.	4. Locate leaks then replace packing.
5. Reservoir too small to provide adequate colling.	5. Replace with larger reservoir, or install cooler.
6. Undersize valves or piping.	6.Check flow velocity through lines and valves and compare with manufactaurers' recommendations. If excessive, replace by installing larger equipment.

Note: If system operates continually at high operating temperatures, consideration should be given to the installation of an oil cooler.

7.5 Improper Operation of Fluid Motors

A. Motor turning in wrong direction

Possible Causes	Remedy
1. Incorrect piping between control valve and motor.	1. Check circuit to determine correct piping.

B. Motor not turning over or not developing proper speed or torque

Possible Causes	Remedy
1.System overload relief valve adjustment not set high enough.	1. Check system pressure and reset relief valve.
2.Relief valve sticking open.	2. Remove dirt under pressure adjustment ball or piston.
3. Free recirculation of oil to reservoir being allowed through system.	3. Directional cntrol valve may be in open center neutral or other return line unintentionally open. Repair or replace valve.

4. Driven mechanism binding because of misalignment.	torque requirement of driven shaft.
5. Pump not delivering sufficient pressure or volume.	5. Check pump delivery and pressure.
6. Motor yoke not set a proper angle (on adjustable motors).	6. Adjust pump yoke angle by means of hand wheel.

C. External oil leakage from motor

Possible Causes	Remedy
1. Gaskets leaking (may be due to reservoir drain not being connected if this is required).	1. Replace. (If drain line required it must be piped directly to reservoir.

REFERENCES

Operation and Care of Hydraulic Machinery. Copyright 1949, 54, 62, 65. Texaco, Inc., 135 East 42nd St., New York, N.Y. 10017.

Service Manual for Heavy Duty Transmissions. Sundstrand Corportion, 2800 East 13th St., Ames, Iowa 50010

Fluid Power-Designers Lightning Reference Handbook. Copyright 1976. Paul-Monroe Hydraulics, Inc., 1701 West Sequoia Avenue, Orange, Ca. 92668. Phone 714-978-9600.

Fluid Power Design Engineers Handbook. Copyright 1973. Parker-Hannifin Corporation, 17325 Euclid Avenue, Dept. HP, Cleveland, Ohio 44112. Phone 216-531-3000.

Hydraulics and Pneumatics Magazine. 614 Superior Ave. West, Cleveland, Ohio 44113. Phone 216-696-0300

Thomas E. Harper. Mechanical Systems Section, U.S. Naval Oceanographic Office, Bay St. Louis, MS 39522. Phone 601-688-4276.

L.G. Galli. Code 5004, Naval Research Laboratory, Washington, D.C. 20390. Phone 202-767-3265.

CHAPTER 12

Useful Information

1.0 WIRE ROPE DATA 12-1
2.0 ELECTRO-MECHANICAL CABLE DATA 12-6
3.0 NYLON ROPE DATA 12-8
4.0 POLYESTER ROPE DATA 12-9
5.0 POLYPROPELENE ROPE DATA 12-10
6.0 MANILA ROPE DATA 12-11
7.0 CHAIN DATA 12-12
8.0 MARINE HARDWARE DATA 12-13
9.0 METRIC CONVERSIONS 12-21
10.0 ENGINEERING UNITS 12-23

Wire Rope Data

Bright or AMGAL MONITOR AA Torque-Balanced Rope

Size Inches	Construction (Seale)	Wt. in Air - lbs/ft	Wt. in Water lbs/ft	Approx. Elastic Limit	Breaking Load lbs.	0.2 % Yield Strength lbs	Max. Length ft
3/16	3x19Seale	.0586	.0509	3,000	4,000	3,500	50,000
1/4	3x19 "	.0997	.0867	3,063	6,750	5,900	45,000
5/16	3x19 "	.153	.133	7,725	10,300	9,100	30,000
3/8	3x19 "	.220	.191	11,100	14,800	13,000	50,000
7/16	3x19 "	.304	.264	15,000	20,000	17,600	42,000
1/2	3x19 "	.392	.341	19,275	25,700	22,600	98,000
9/16	3x19 "	.492	.428	24,375	32,500	28,600	77,000
5/8	3x19 "	.602	.523	30,225	40,300	35,500	62,000
3/4	3x19 "	.879	.764	43,350	57,800	50,900	43,000
7/8	3x19 "	1.21	1.05	58,500	78,000	68,600	32,000
1	3x19 "	1.56	1.36	75,450	100,600	88,500	24,000
1 1/8	3x19 "	1.96	1.70	93,000	124,000	109,000	19,000
	Seale FW						
1/2	3x46	.417	.362	19,275	25,700	22,600	98,000
9/16	3x46	.517	.449	24,375	32,500	28,600	77,000
5/8	3x46	.631	.548	30,225	40,300	35,500	62,000
3/4	3x46	.903	.785	43,350	57,800	50,900	43,000
7/8	3x46	1.27	1.10	58,500	78,000	68,600	32,000
1	3x46	1.64	1.43	75,450	100,600	88,500	24,000
1 1/8	3x46	2.07	1.80	93,000	124,000	109,000	19,000
1 1/4	3x46	2.60	2.26	118,500	158,000	139,000	15,500
1 3/8	3x46	3.10	2.69	141,000	188,000	165,000	12,900
1 1/2	3x46	3.69	3.21	166,500	222,000	195,000	10,800
1 5/8	3x46	4.43	3.85	198,750	265,000	233,000	9,200
1 3/4	3x46	5.12	4.45	228,000	304,000	267,000	8,000

* Data Courtesy of US Steel

3 x 7 TYPE 304 STAINLESS STEEL ROPE

Size	Minimum Breaking Strength lbs.	Approx. Elastic Limit lbs.	Min. 0.2% Yield Strength lbs.	Weight lbs./ft.	Area Sq. In.
5/32	2,800	2,100	2,460	.0406	.01096
11/64	3,300	2,470	2,900	.0491	.01326
3/16	3,900	2,920	3,500	.0578	.01561
7/32	5,000	3,750	4,500	.0745	.02011

3 x 19 TYPE 304 STAINLESS STEEL ROPE

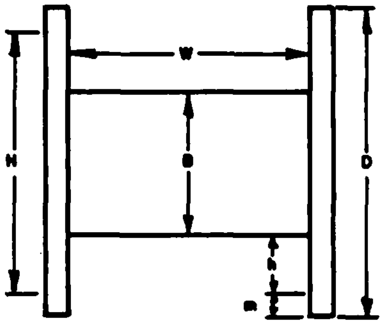
11/64	3,500	2,620	3,100	.0512	.01394
3/16	4,000	3,000	3,500	.0592	.01611
7/32	5,400	4,050	4,750	.0803	.02184

3 x 19 TENELON STAINLESS STEEL
(*for non-magnetic applications)

3/8	12,700	8,900	11,200	.221	.06015
7/16	17,200	12,900	15,100	.299	.08114
1/2	22,000	16,500	19,400	.388	.10529
9/16	28,000	21,000	24,100	.487	.13255

* This product for non-magnetic applications. To improve its resistance to Stress Corrosion, the wires may be galvanized.

- Data Courtesy of US Steel -



Let D = Diameter of Head in Inches.
B = Diameter of Barrel in Inches.
h = Depth of Cable in Inches.
W = Width between Flanges in Inches.
d = Diameter of Cable in Inches.
L = Length of Cable in Feet.
m = margin.

To Compute Length of Cable in Feet for any Reel or Drum:
$L = \text{Factor} \times W \times h \times (B + h)$
A table of factors for ropes to the maximum oversize tolerance (as shown previously in this handbook) is presented below.

Nominal Rope Dia.	Factor	Nominal Rope Dia.	Factor
1/4	3.728	2 1/4	.0469
5/16	2.432	2 3/8	.0421
3/8	1.689	2 1/2	.0380
7/16	1.241	2 5/8	.0345
1/2	.9498	2 3/4	.0314
9/16	.7505	2 7/8	.0287
5/8	.6079	3	.0264
3/4	.4222	3 1/8	.0243
7/8	.3102	3 1/4	.0225
1	.2375	3 3/8	.0208
1 1/8	.1876	3 1/2	.0194
1 1/4	.1520	3 5/8	.0181
1 3/8	.1256	3 3/4	.0169
1 1/2	.1055	3 7/8	.0158
1 5/8	.0899	4	.0148
1 3/4	.0775	4 1/4	.0131
1 7/8	.0675	4 1/2	.0117
2	.0594	4 3/4	.0105
2 1/8	.0526	5	.0095

The Formula can be readily derived:

(1) Length of Coil of Middle Layer

$$= \frac{\pi}{12}\left(B + \frac{H - B}{2}\right)$$

Number of Coils $= \frac{W}{d}$

Number of Layers $= \frac{H - B}{2d}$

$$L = \frac{\pi}{12}\left(B + \frac{H - B}{2}\right) \times \frac{W}{d} \times \frac{H - B}{2d}$$

$$= \frac{\pi W(H + B)(H - B)}{48d^2}$$

(2) Volume of Drum in Cubic Inches

$$= W\left(\frac{\pi H^2}{4} - \frac{\pi B^2}{4}\right)$$

$$L = \frac{W}{12d^2}\left(\frac{\pi H^2 - \pi B^2}{4}\right) = \frac{\pi W}{48d^2}(H^2 - B^2)$$

$$= \frac{\pi W(H + B)(H - B)}{48d^2}$$

$$L = \frac{\pi W(H + B)(H - B)}{48d^2} = \frac{W(H+B)(H-B)}{15.28d^2}$$

$$= \frac{.06545\ W(H + B)(H - B)}{d^2}$$

$$= \frac{.2618\ Wh\ (B + h)}{d^2}$$

$$\text{Let Factor} = \frac{.2618}{d^2}$$

then

$$L = \text{Factor} \times W \times h \times (B + h)$$

When the diameter of the rope is not full oversize or when strand is to be reeled, the actual product diameter should be used with the formula

$$L = \frac{.2618\ Wh\ (B + h)}{d^2}$$

to determine capacities.

When shipping rope on reels, the reels should not be completely filled. A margin (m) should be left to protect the rope.

This Formula is based on the assumption that: the rope is oversize and does not flatten when coiled; and that it is in perfectly uniform layers with no meshing of the coils. These factors vary with size and construction of the cable and with the dimensions of the reel or drum. As these variables tend to offset each other, this method of computing reel and drum capacities has proved to be reliable.

ELECTRO-MECHANICAL CABLE DATA
MECHANICAL CHARACTERISTICS

TYPE	O.D.	CORE DIA.	JACKET THICKNESS	LB. BREAK STRENGTH	WT./1000' WT. AIR	WT./1000' WT. WATER	MIN. SHEAVE DIAMETER	CALC. ARMOR RATIO
SINGLE CONDUCTOR								
1-H-100	.102	----	----	1000	19.	15.	6	2.2
1-H-112	.112	----	----	1300	21.4	17.	5	1.9
1-H-142	.112	----	.015 PE	1300	23.9	17.5	5	1.7
1-H-150	.112	----	.019 PE	1000	20.	12.9	5	1.17
1-H-169	.169	----	----	2400	46.	37.	7	1.9
1-H-225	.252	----	----	5600	112.	90.	14	2.1
1-H-291	.291	----	----	8800	153.	127.	14	1.4
COAXIAL CABLES								
2-H-161	.157	.094	----	1300	46.	38.	7	1.9
2-H-188	.185	.096	- ----	2900	58.	47.	10	2.3
2-H-251	.251	.146	----	4900	104.	84.	12	2.3
2-H-252	.252	.133	----	5500	107.	87.	14	2.3
2-H-255	.255	.146	----	5000	107.	86.	11	1.4
2-H-377	.377	.226	----	10,000	212.	167.	17	1.4
2-H-470	.295	.161	.058 Pb	6,400	493.	416.	16	2.4
2-H-477	.477	.285	----	17,000	354.	282.	22	1.25
2-H-528	.528	.304	----	20,000	434.	346.	22	1.5
2-H-609	.609	.404	----	26,000	514.	384.	24	1.06
2-H-678	.670	.418	----	33,000	700.	559.	28	1.9
2-H-683	.680	.450	----	31,000	659.	514.	26	1.04
2-H-696	.698	.350	----	45,000	824.	671.	27	1.05
2-H-726	.726	.470	----	40,000	764.	398.	30	1.07

*Courtesy Rochester Corporation

ELECTRO-MECHANICAL CABLE DATA

MECHANICAL CHARACTERISTICS

*Courtesy Rochester Corporation

TYPE	O.D.	Core Dia.	Lb Break Strength	WT/1000' Wt Air	WT/1000' Wt Water	Min. Sheave Diameter	Calc. Armor Ratio
MULTI-CONDUCTOR CABLES							
3-H-182	.183	.098	2,900	55	44	10	2.3
3-H-187	.191	.144	2,900	57	47	9	2.3
3-H-250	.254	.134	5,500	109	88	14	2.5
3-H-292	.292	.159	7,200	136	109	15	2.6
3-H-305	.305	.177	7,400	145	116	13	1.6
3-H-322	.321	.170	9,200	170	137	17	2.5
3-H-419	.421	.219	16,000	308	252	23	2.5
3-H-420	.419	.262	12,500	274	219	18	1.9
3-H-460	.450	.268	20,500	312	246	20	1.09
4-H-185	1.83	.098	2,900	57	46	10	2.6
4-H-225	.224	.120	4,400	82	67	12	2.5
4-H-250	.254	.134	5,500	103	81	14	2.5
4-H-292	.282	.149	7,200	132	107	16	2.6
4-H-349	.349	.201	9,700	194	156	15	1.6
4-H-350	.351	.185	10,400	213	174	19	2.7
4-H-375	.375	.199	11,800	219	175	20	2.6
7-H-325	.323	.172	9,200	174	141	18	2.3
7-H-374	.375	.199	11,800	240	193	20	2.6
7-H-420	.420	.262	12,500	278	222	18	1.9
7-H-422	.421	.219	16,500	305	249	23	2.5
7-H-463	.464	.289	16,000	324	255	20	2.0
8-H-575	.589	.389	23,000	499	380	23	1.2
10-H-465	.463	.239	16,000	312	244	20	2.0
12-H-675	.671	.417	30,000	667	521	28	1.8
37-H-10	1.023	.661	72,000	1,508	1,178	42	1.13

NYLON ROPE

Dia.	Cir.	Pounds/ 100 Feet	Feet/ Pound	New Rope Tensile Strength (lb.)	Working Load (lb.)
3/16	5/8	1.0	100.0	900	75
1/4	3/4	1.5	66.7	1,490	124
5/16	1	2.5	40.0	2,300	192
3/8	1 1/8	3.5	28.5	3,350	278
7/16	1 1/4	5.0	20.0	4,500	410
1/2	1 1/2	6.5	15.4	5,750	525
9/16	1 3/4	8.3	12.3	7,200	720
5/8	2	10.5	9.5	9,350	935
3/4	2 1/4	14.5	6.9	12,800	1,420
13/16	2 1/2	17.0	5.9	15,300	1,700
7/8	2 3/4	20.0	5.0	18,000	2,000
1	3	26.0	3.8	22,500	2,500
1 1/16	3 1/4	29.0	3.4	25,900	2,880
1 1/8	3 1/2	34.0	2.9	29,700	3,320
1 1/4	3 3/4	40.0	2.5	33,750	3,760
1 5/16	4	45.0	2.2	38,750	4,320
1 1/2	4 1/2	55.0	1.8	47,700	5,320
1 5/8	5	68.0	1.5	58,500	6,500
1 3/4	5 1/2	83.0	1.2	70,200	7,800
2	6	95.0	1.05	82,800	9,200
2 1/8	6 1/2	109.0	0.92	95,400	10,600
2 1/4	7	129.0	0.77	113,000	12,600
2 1/2	7 1/2	149.0	0.67	126,000	14,000
2 5/8	8	168.0	0.59	146,000	16,200
2 7/8	8 1/2	189.0	0.53	162,000	18,000
3	9	210.0	0.47	180,000	20,000
3 1/4	10	263.0	0.38	225,000	25,200
3 1/2	11	316.0	0.32	270,000	30,000
4	12	379.0	0.26	324,000	36,000

*Courtesy The Columbian Group

POLYESTER ROPE

Dia.	Cir.	Pounds/ 100 Feet	Feet Pound	New Rope Tensile Strength (lb.)	Working Load (lb.)
3/16	5/8	1.2	83.4	900	90
1/4	3/4	2.0	50.0	1,490	149
5/16	1	3.1	32.2	2,300	230
3/8	1 1/8	4.5	22.2	3,350	334
7/16	1 1/4	6.2	16.1	4,500	500
1/2	1 1/2	8.0	12.5	5,750	640
9/16	1 3/4	10.2	9.8	7,200	900
5/8	2	13.0	7.7	9,000	1,130
3/4	2 1/4	17.5	5.7	11,300	1,610
13/16	2 1/2	21.0	4.8	14,000	2,000
7/8	2 3/4	25.0	4.0	16,200	2,320
1	3	30.5	3.3	19,800	2,820
1 1/16	3 1/4	34.5	2.9	23,000	3,280
1 1/8	3 1/2	40.0	2.5	26,600	3,800
1 1/4	3 3/4	46.3	2.2	29,900	4,260
1 5/16	4	52.5	1.9	33,800	4,820
1 1/2	4 1/2	66.8	1.5	42,100	6,050
1 5/8	5	82.0	1.2	51,300	7,350
1 3/4	5 1/2	98.0	1.02	61,000	8,700
2	6	118.0	0.85	72,000	10,300
2 1/8	6 1/2	135.0	0.74	82,800	11,900
2 1/4	7	157.0	0.64	96,300	13,800
2 1/2	7 1/2	181.0	0.55	110,000	15,700
2 5/8	8	205.0	0.49	123,000	17,600
2 7/8	8 1/2	230.0	0.43	139,000	19,900
3	9	258.0	0.39	157,000	22,400
3 1/4	10	318.0	0.31	189,000	27,000
3 1/2	11	384.0	0.26	229,000	32,600
4	12	460.0	0.22	270,000	38,600

*Courtesy The Columbian Group

POLYPROPELENE ROPE

Dia.	Cir.	Pounds/ 100 Feet	Feet/ Pound	New Rope Tensile Strength (lb.)	Working Load (lb.)
3/16	5/8	.70	143.0	720	72
1/4	3/4	1.2	83.4	1,130	113
5/16	1	1.8	55.6	1,710	171
3/8	1 1/8	2.8	35.7	2,430	244
7/16	1 1/4	3.8	26.3	3,150	352
1/2	1 1/2	4.7	21.3	3,780	420
9/16	1 3/4	6.1	16.4	4,590	575
5/8	2	7.5	13.3	5,580	700
3/4	2 1/4	10.7	9.3	7,650	1,090
13/16	2 1/2	12.7	7.9	8,910	1,270
7/8	2 3/4	15.0	6.7	10,400	1,490
1	3	18.0	5.5	12,600	1,800
1 1/16	3 1/4	20.4	4.9	14,400	2,060
1 1/8	3 1/2	23.7	4.2	16,500	2,360
1 1/4	3 3/4	27.0	3.7	18,900	2,700
1 5/16	4	30.5	3.3	21,200	3,020
1 1/2	4 1/2	38.5	2.6	26,700	3,820
1 5/8	5	47.5	2.1	32,400	4,620
1 3/4	5 1/2	57.0	1.7	38,700	5,550
2	6	69.0	1.4	46,800	6,700
2 1/8	6 1/2	80.0	1.2	54,900	7,850
2 1/4	7	92.0	1.1	62,100	8,850
2 1/2	7 1/2	107.0	0.93	72,000	10,300
2 5/8	8	120.0	0.83	81,000	11,600
2 7/8	8 1/2	137.0	0.73	90,900	13,000
3	9	153.0	0.65	103,000	14,700
3 1/4	10	190.0	0.53	123,000	17,600
3 1/2	11	232.0	0.43	146,000	20,800
4	12	275.0	0.36	171,000	24,400

*Courtesy The Columbian Group

MANILA

Regular Construction

Dia.	Cir.	Pounds/ 100 Feet	Feet Pound	New Rope Tensile Strength (lb.)	Working Load (lb.)
3/16	5/8	1.5	66.60	405	41
1/4	3/4	2.0	50.0	540	54
5/16	1	2.9	34.50	900	90
3/8	1 1/8	4.1	24.40	1,215	122
7/16	1 1/4	5.3	19.00	1,575	176
1/2	1 1/2	7.5	13.33	2,385	264
9/16	1 3/4	10.4	9.61	3,105	388
5/8	2	13.3	7.50	3,960	496
3/4	2 1/4	16.7	6.00	4,860	695
13/16	2 1/2	19.5	5.13	5,850	835
7/8	2 3/4	22.5	4.45	6,930	995
1	3	27.0	3.71	8,100	1,160
1 1/16	3 1/4	31.3	3.20	9,450	1,350
1 1/8	3 1/2	36.0	2.78	10,800	1,540
1 1/4	3 3/4	41.8	2.40	12,150	1,740
1 5/16	4	48.0	2.09	13,500	1,930
1 1/2	4 1/2	60.0	1.67	16,650	2,380
1 5/8	5	74.4	1.34	20,250	2,880
1 3/4	5 1/2	89.5	1.12	23,850	3,400
2	6	108.0	.93	27,900	4,000
2 1/8	6 1/2	125.0	.79	32,400	4,620
2 1/4	7	146.0	.685	36,900	5,300
2 1/2	7 1/2	167.0	.59	41,850	5,950
2 5/8	8	191.0	.52	46,800	6,700
2 7/8	8 1/2	215.0	.47	52,200	7,400
3	9	242.0	.42	57,600	8,200
3 1/4	10	299.0	.33	69,300	9,950
3 1/2	11	367.0	.27	81,900	11,700
4	12	436.0	.23	94,500	13,500

*Courtesy The Columbian Group

CHAIN

ENGINEERING SPECIFICATIONS

CROSBY PROOF COIL - SPECTRUM 3 CHAIN

Trade Size Inches	Size Material Inches	Working Load Limit/lb.	Nominal Inside Length/in.	Nominal Inside Width/in.	Max.(in) Length 100Links	Wt./100 ft.in lbs.
3/16	.218	850	.95	.40	99	40
1/4	.281	1,450	1.00	150	104	73
5/16	.343	2,200	1.10	.50	114	110
3/8	.406	3,050	1.23	.62	128	159
1/2	.531	5,150	1.50	.81	156	275
5/8	.656	7,900	1.87	1.00	194	408
3/4	.781	11,150	2.12	1.12	220	581

CROSBY HIGH TEST - SPECTRUM 4 CHAIN

1/4	.281	2,600	.82	.39	86	77
5/16	.343	3,900	1.01	.48	105	117
3/8	.406	5,400	1.15	.56	121	165
7/16	.468	7,200	1.29	.65	135	220
1/2	.531	9,200	1.43	.75	150	262
5/8	.656	12,750	1.79	.90	186	422
3/4	.781	18,500	1.96	1.06	205	615

CROSBY SPECTRUM 7 CHAIN

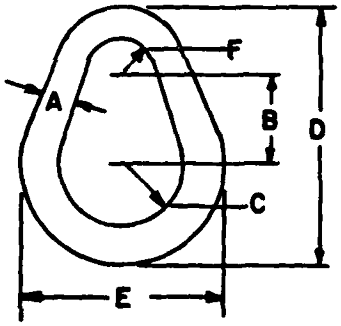
1/4	.281	3,600	.82	.39	86	77
5/16	.343	5,400	1.01	.48	105	117
3/8	.406	7,500	1.15	.56	121	165
7/16	.468	10,000	1.29	.65	135	220
1/2	.531	12,750	1.43	.75	150	262
5/8	.656	19,000	1.79	.90	186	422

CROSBY ALLOY - SPECTRUM 8 CHAIN

					(Links Per/ft)	
1/4	.280	4,100	.85	.42	14 1/8	75
5/16	.343	5,100	1.00	.47	12	92
3/8	.390	7,300	1.14	.53	10 1/2	145
1/2	.515	13,000	1.43	.69	8 3/8	256
5/8	.640	20,300	1.74	.83	6 7/8	404
3/4	.765	29,300	2.04	.97	5 7/8	575
7/8	.875	39,900	2.32	1.11	5 1/8	730

*Courtesy of Crosby Group

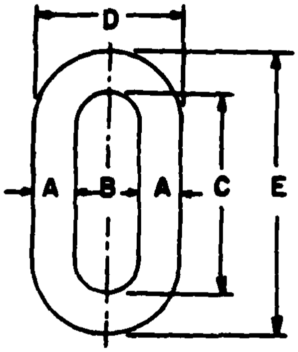
SLING LINK



STOCK DIA. A	B	C	D	E	F	WEIGHT EACH	SAFE LOAD* SINGLE PULL POUNDS
3/8	1.13	.75	3.00	2.25	.38	.23	1,800
1/2	1.50	1.00	4.00	3.00	.50	.53	2,900
5/8	1.875	1.25	5.00	3.75	.63	1.1	4,200
3/4	2.25	1.50	6.00	4.50	.75	1.9	6,000
7/8	2.63	1.75	7.00	5.25	.88	2.9	8,300
1	3.00	2.00	8.00	6.00	1.00	4.3	10,800
1 1/4	4.00	2.50	10.25	7.50	1.25	8.5	16,750
1 3/8	4.13	2.75	11.00	8.25	1.38	11.3	20,500

*Minimum ultimate strength six times safe working load.

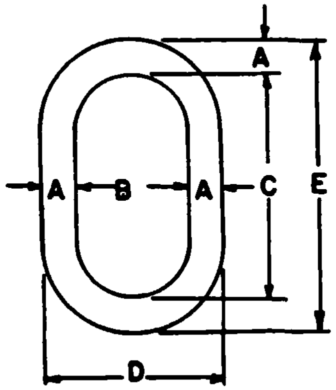
END LINK



STOCK DIA. A	B	C	D	WEIGHT EACH	SAFE LOAD* POUNDS
5/16	.50	1.75	1.13	.14	2,500
3/8	.56	1.88	1.31	.22	3,800
1/2	.75	2.38	1.75	.48	6,500
5/8	1.00	3.25	2.13	.92	9,300
3.4	1.13	3.50	2.63	1.37	14,000
7/8	2.00	5.13	3.75	2.75	12,000 .
1	2.25	5.75	4.25	3.6	15,200
1 1/4	2.50	7.00	5.00	7	26,400
1 3/8	2.75	7.75	5.50	10	30,000

*Ultimate Load Five Times Safe Working Load.

MASTER LINK

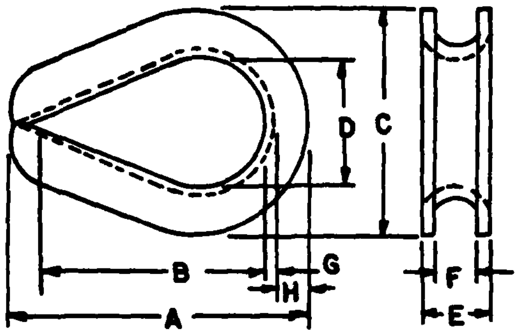


STOCK DIA. A	B	C	D	E	WEIGHT EACH	SAFE LOAD* SINGLE PULL POUNDS
1/2	2.50	5.00	3.25	6.00	.81	3,250
5/8	3.00	6.00	4.25	7.25	1.5	4,400
3/4	2.75	5.50	4.25	7	2	7,000
1	3.50	7.00	5.50	9	4.6	16,500
1 1/4	4.38	8.75	6.88	11.25	9.2	25,000
1 1/2	5.25	10.50	8.25	13.50	15.7	35,500
1 3/4	6.00	12.00	9.50	15.50	24.5	44,500
2	7.00	14.00	11.00	18.00	38.1	57,500
† 2 1/4	8.00	16.00	12.50	20.50	54.8	67,000
† 2 3/4	9.50	16.00	15.00	21.50	87.7	100,000

* Minimum Ultimate Strength Six times safe working load.
† Contact Crosby Group Office on Availability.

Courtesy The Crosby Group

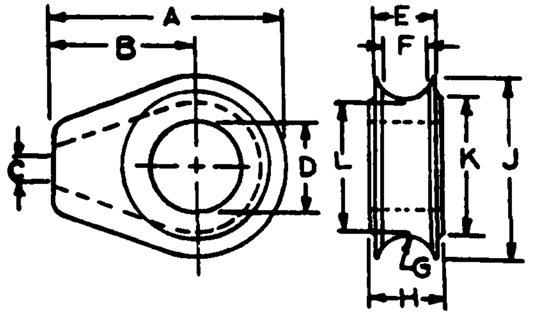
HEAVY WIRE ROPE THIMBLES



ROPE DIA.	A	B	C	D	E	F	G	H	WT. PER 100
1/4	2.19	1.63	1.50	.88	.41	.28	.06	.23	.5
5/16	2.50	1.88	1.81	1.06	.50	.34	.08	.28	14
3/8	2.88	2.13	2.13	1.13	.63	.41	.11	.34	25
7/16	3.25	2.38	2.38	1.25	.72	.47	.13	.38	36
1/2	3.63	2.75	2.75	1.50	.81	.53	.14	.41	51
9/16	3.63	2.75	2.69	1.50	.88	.59	.14	.41	51
5/8	4.25	3.25	3.13	1.75	.97	.66	.16	.50	75
3/4	5.00	3.75	3.81	2.00	1.22	.78	.22	.66	147
7/8	5.50	4.25	4.25	2.25	1.38	.94	.22	.75	185
1	6.13	4.50	4.94	2.50	1.56	1.06	.25	.88	300
1 1/8 - 1 1/4	7.00	5.13	5.88	2.88	1.81	1.31	.25	1.13	410
1 1/4 - 1 3/8	9.06	6.50	6.81	3.50	2.19	1.44	.38	1.13	834
1 3/8 - 1 1/2	9.00	6.25	7.13	3.50	2.56	1.56	.50	1.13	1200
1 5/8	11.25	8.00	8.13	4.00	2.72	1.72	.50	1.38	1625
1 3/4	12.19	9.00	8.50	4.50	2.84	1.84	.50	1.31	1800
1 7/8 - 2	15.13	12.00	10.38	6.00	3.09	2.09	.50	1.50	2600
2 1/4	17.13	14.00	11.88	7.00	3.63	2.38	.63	1.63	4300

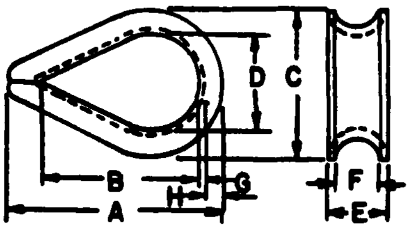
*Courtesy The Crosby Group

SOLID WIRE ROPE THIMBLES



ROPE DIA.	A	B	C	D	E	F	G	H	J	K	L	WT.EA. LBS.
1/2	2.81	1.75	.25	1.06	.75	.56	.28	.88	2.13	1.63	1.56	1
5/8	4.69	3.00	.38	1.31	1.06	.81	.41	1.13	3.38	2.25	2.56	2.5
3/4	4.69	3.00	.38	1.50	1.06	.81	.41	1.38	3.38	2.25	2.56	3.3
7/8	6.06	3.81	.50	1.75	1.38	1.06	.53	1.63	4.50	3.25	3.44	5
1	6.06	3.81	.50	2.13	1.38	1.06	.53	1.81	4.50	3.25	3.44	6.5
1 1/8	7.25	4.56	.63	2.38	1.75	1.31	.66	2.06	5.38	3.88	4.06	8.5
1¼ - 1 3/8	7.25	4.56	.63	2.63	1.94	1.53	.78	2.31	5.38	3.88	4.13	10

STANDARD WIRE ROPE THIMBLES



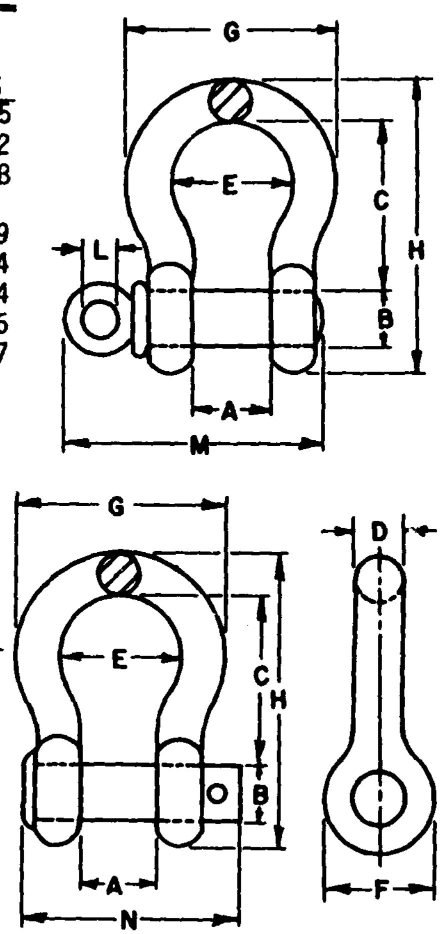
ROPE DIA.	A	B	C	D	E	F	G	H	WT.PER 100
1/8	1.94	1.31	1.06	.69	.25	.16	.05	.13	3.3
3/16	1.94	1.31	1.06	.69	.31	.22	.05	.13	3.3
1/4	1.94	1.31	1.06	.69	.38	.28	.05	.13	3.3
5/16	2.13	1.50	1.25	.81	.44	.34	.05	.13	4
3/8	2.38	1.63	1.47	.94	.53	.41	.06	.16	7.5
1/2	2.75	1.88	1.75	1.13	.69	.53	.08	.19	13.8
5/8	3.50	2.25	2.38	1.38	.91	.66	.13	.34	36
3/4	3.75	2.50	2.69	1.63	1.08	.78	.14	.34	50
7/8	5.00	3.50	3.19	1.88	1.27	.94	.16	.44	90
1	5.69	4.25	3.75	2.50	1.39	1.06	.16	.41	105
1 1/8 - 1 1/4	6.25	4.50	4.31	2.75	1.75	1.31	.22	.50	186
1 1/2	7.50	5.00	5.38	3.25	2.06	1.56	.25	.75	340
1 3/4	10.75	7.25	7.25	4.25	2.38	1.88	.25	1.19	594
2	10.75	7.25	7.25	4.25	2.63	2.13	.25	1.19	594

*Courtesy The Crosby Group

SCREW PIN ANCHOR SHACKLES

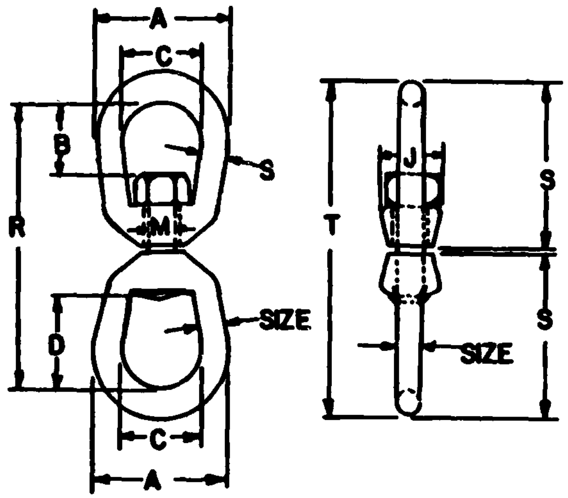
S.W.L. TONS	SIZE D	A	B	C	E	F	G	H	L	M	N	TOLERANCE + or - C	TOLERANCE + or - A	WT. EA. LBS
1/3	.19	.38	.25	.88	.69	.56	.98	1.47	.13	1.13	--	.06	.06	.05
1/2	.25	.50	.31	1.13	.78	.69	1.28	1.88	.16	1.44	1.34	.06	.06	.12
3/4	.31	.53	.38	1.22	.84	.81	1.47	2.13	.19	1.72	1.59	.06	.06	.18
1	.38	.66	.44	1.44	1.03	.97	1.78	2.53	.22	2.06	1.88	.13	.06	.3
1 1/2	.44	.72	.50	1.69	1.16	1.06	2.03	2.91	.25	2.34	2.13	.13	.06	.49
2	.50	.81	.63	1.88	1.31	1.19	2.31	3.28	.31	2.72	2.38	.13	.06	.74
3 1/4	.63	1.06	.75	2.38	1.69	1.56	2.94	4.22	.38	3.41	2.91	.13	.06	1.44
4 3/4	.75	1.25	.88	2.81	2.00	1.88	3.50	5	.44	4.03	3.44	.25	.06	2.16
6 1/2	.88	1.44	1.00	3.31	2.28	2.13	4.03	5.75	.50	4.63	3.84	.25	.06	3.37
8 1/2	1.00	1.69	1.13	3.75	2.69	2.38	4.69	6.50	.56	5.31	4.53	.25	.06	5.3
9 1/2	1.13	1.81	1.25	4.25	2.91	2.63	5.16	7.31	.63	5.88	5.13	.25	.06	7
12	1.25	2.03	1.38	4.69	3.25	3.00	5.75	8.13	.69	6.44	5.50	.25	.06	9.6
13 1/2	1.38	2.25	1.50	5.25	3.63	3.31	6.38	9.03	.75	7.13	6.13	.25	.13	12.6
17	1.50	2.38	1.63	5.75	3.88	3.63	6.88	9.88	.81	7.66	6.50	.25	.13	17.3
25	1.75	2.88	2.00	7.00	5.00	4.31	8.50	11.88	1.00	9.19	7.75	.25	.13	27.8
35	2.00	3.25	2.25	7.75	5.75	5.00	9.75	13.38	1.13	10.34	8.75	.25	.13	41.1
50	2.50	4.13	2.75	10.50	7.25	6.00	12.25	17.38	1.38	12.97	10.94	.75	.13	83.5
75	3.00	5.00	3.25	13.00	7.88	6.50	13.88	20.88	--	--	13.25	.75	.13	119

Proof Load 2.2 times safe working load.
Ultimate Strength 6 times safe working load through 50 ton capacity.
Ultimate Strength 5 times safe working load on 75,100 and 130 ton capacity.



*Courtesy The Crosby Group

REGULAR SWIVEL

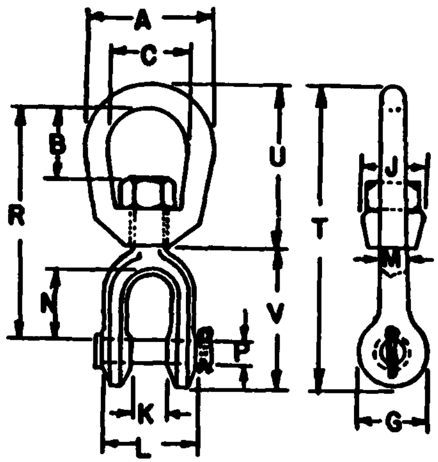


SIZE	SAFE WORK LOAD POUNDS	A	B	C	D	J	M	R	S	T	WEIGHT LBS/EA.
1/4	850	1.25	.69	.75	1.06	.69	.31	2.94	1.69	3.44	.19
5/16	1,250	1.63	.81	1.00	1.25	.81	.38	3.56	2.06	4.19	.31
3/8	2,250	2.00	.94	1.25	1.50	1.00	.50	4.31	2.50	5.06	.68
1/2	3,600	2.50	1.31	1.50	2.00	1.31	.63	5.44	3.19	6.44	1.25
5/8	5,200	3.00	1.56	1.75	2.38	1.50	.75	6.56	3.88	7.81	2.25
3/4	7,200	3.50	1.75	2.00	2.63	1.88	.88	7.19	4.31	8.69	3.5
7/8	10,000	4.00	2.06	2.25	3.06	2.13	1.00	8.38	5.00	10.13	5.4
1	12,500	4.50	2.31	2.50	3.50	2.38	1.13	9.63	5.75	11.63	8.8
1 1/8	15,200	5.00	2.38	2.75	3.75	2.56	1.25	10.38	6.25	12.63	12
1 1/4	18,000	5.63	2.69	3.13	3.69	3.00	1.38	11.13	6.75	13.63	16
1 1/2	45,200	7.00	4.19	4.00	4.19	4.00	2.25	17.13	10.00	20.13	19

Ultimate Tensile Strength Five Times Safe Working Load.

*Courtesy The Crosby Group

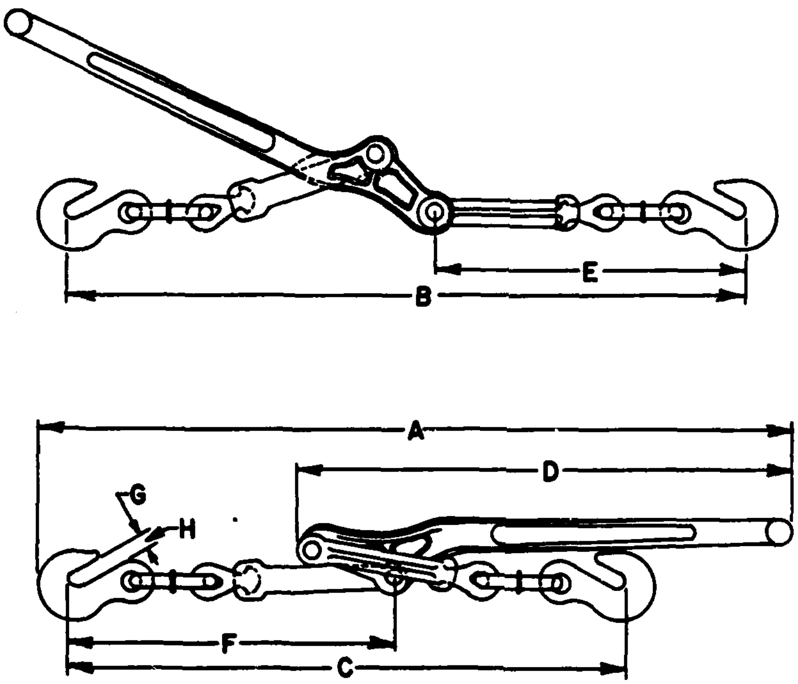
JAW END SWIVEL



SIZE	SAFE WORK LOAD LBS.	A	B	C	G	J	K	L	M	N	P	R	T	U	V	WT.LBS EACH
1/4	850	1.25	.69	.75	.69	.69	.47	1.03	.31	.88	.25	2.63	3.38	1.69	1.69	.22
5/16	1,250	1.63	.81	1.00	.81	.81	.50	1.13	.38	.88	.31	2.94	3.88	2.06	1.81	.31
3/8	2,250	2.00	.94	1.25	1.00	1.00	.63	1.41	.50	1.06	.38	3.63	4.75	2.50	2.25	.56
1/2	3,600	2.50	1.31	1.50	1.31	1.31	.75	1.75	.63	1.31	.50	4.50	6.06	3.19	2.88	1.25
5/8	5,200	3.00	1.56	1.75	1.63	1.50	.94	2.06	.75	1.50	.63	5.31	7.31	3.88	3.44	2.13
3/4	7,200	3.50	1.75	2.00	1.88	1.88	1.13	2.53	.88	1.75	.75	6.06	8.31	4.31	4.00	3.5
7/8	10,000	4.00	2.06	2.25	2.13	2.13	1.19	2.75	1.00	2.06	.88	7.00	9.53	5.00	4.53	5.3
1	12,500	4.50	2.31	2.50	2.63	2.38	1.75	3.72	1.13	2.81	1.13	8.56	11.69	5.75	5.94	9.8
1 1/8	15,200	5.00	2.38	2.75	2.63	2.56	1.75	3.72	1.25	2.81	1.13	8.84	12.19	6.25	5.94	14
1 1/4	18,000	5.63	2.69	3.13	3.13	3.00	2.06	4.31	1.50	2.81	1.38	9.44	13.13	6.75	6.38	17
1 1/2	45,200	7.00	4.19	4.00	5.63	4.00	2.88	6.00	2.25	4.44	2.25	14.74	20.84	10.0	10.84	49

Ultimate Tensile Strength Five Times Safe Working Load.

*Courtesy The Crosby Group



MODEL	A	B	C	D	E	F	G	H CHAIN SIZE	TAKE UP
7-1	24.19	21.5	17.25	16.0	10.0	10.5	0.5	5/16-3/8	4.25
A-1	28.0	24.63	20.13	18.5	11.63	11.56	0.63	3/8 -1/2	4.5
C-1	30.5	28.25	23.5	19.5	13.5	13.25	0.75	1/2 -5/8	4.75
L-1	35.75	31.0	26.5	23.25	15.5	14.13	0.94	5/8 -3/4	4.5

LOAD BINDER

*Courtesy The Crosby Group

- METRIC CONVERSIONS -

METRIC EQUIVALENTS OF STANDARD ROPE SIZES

DIAMETER	
Inches	Millimeters
1/64	.40
1/32	.79
3/64	1.2
1/16	1.6
5/64	2.0
3/32	2.4
7/64	2.8
1/8	3.2
5/32	4.0
3/16	4.8
7/32	5.6
1/4	6.3
5/16	7.9
3/8	9.5
7/16	11.1
1/2	12.7
9/16	14.3
5/8	15.9
11/16	17.5
3/4	19.0
13/16	20.6
7/8	22.2
15/16	23.8
1	25.4
1 1/16	27.0
1 1/8	28.6
1 3/16	30.2
1 1/4	31.7
1 3/8	34.9
1 7/16	36.5
1 1/2	38.1
1 5/8	41.3
1 11/16	42.9
1 3/4	44.4
1 13/16	46.0
1 7/8	47.6
1 15/16	49.2
2	50.8
2 1/16	52.4
2 1/8	54.0
2 1/4	57.1
2 1/2	63.5
2 3/4	69.9
3	76.2
3 1/4	82.5
3 1/2	88.9

METRIC EQUIVALENTS OF WEIGHT IN TONS

SHORT TONS	SHORT TONS
1	.90
2	1.81
3	
4	2.72
5	3.62
6	5.44
7	6.35
8	7.25
9	8.16
10	9.07
11	9.97
12	10.88
14	12.70
15	13.60
20	18.14
25	22.67
30	27.21
35	31.75
40	36.28
45	40.82
50	45.35
55	49.89
60	54.43
65	58.96
70	63.50
75	68.03
80	72.57
90	81.64
100	90.75
115	104.32
125	113.39
130	117.93
135	122.47
140	127.00
150	136.07
165	149.68
200	181.43
250	225.79
265	240.40
300	272.15
325	294.83
350	317.51

METRIC CONVERSIONS

ENGLISH TO METRIC

To Find	Multiply	By
Microns	mils	25.4
centimeters	inches	2.54
meters	feet	0.3048
meters	yards	0.9144
kilometers	miles	1.609344
gram	ounces	28.349523
kilogram	pounds	4.5359237×10^{-1}
liters	gallons (U.S.)	3.7854118
liters	gallons (Imp.)	4.546090
mililiters (cc)	fl. ounces	29.573530
sq. centimeters	sq. inches	6.4516
sq. meters	sq. feet	9.290304×10^{-2}
sq. meters	sq. yards	8.3612736×10^{-1}
mililiters (cc)	cu. inches	16.387064
cu. meters	cu. feet	2.8316847×10^{-2}
cu. meters	cu. yards	7.6455486×10^{-1}

Temperature conversion

$$^{o}F = 9/5\ (^{o}C) + 32$$

$$^{o}C = 5/9\ (^{o}F - 32)$$

ENGINEERING UNITS

1 Horsepower	=	33,000 foot pounds per minute 550 foot pounds per second 746 watts .746 kilowatts
1 Horsepower Hour	=	.746 kilowatt hours 1,980,000 foot pounds 2,545 heat units (B.T.U.)
1 Kilowatt	=	1,000 watts 1.34 horsepower 737.3 foot pounds per second 44,240 foot pounds per minute 56.9 heat units (B.T.U.) per minute
1 Kilowatt Hour	=	1,000 watt hours 1.34 horsepower hours 2,654,200 foot pounds 3,412 heat units (B.T.U.)
1 British Thermal Unit	=	1,055 watt seconds 778 foot pounds .000293 kilowatt hour .000393 horsepower hour
1 Watt	=	1 joule per second .00134 horsepower 3,412 heat units (B.T.U.) per hour .7373 foot pounds per second 44.24 foot pounds per minute

INDEX

A

Abrasion, 2-7, 2-9, 2-19
Advanced Cable Dynamics, 7-23
Armor Wires, 2-13, 2-14, 2-24, 2-51, 2-53
Assembly, terminations, 6-4

B

Bearing pressure, sheave, 2-69, 2-70
Buoyancy, E.M., 2-20
Braiding, 2-23
Break strength, 1-2, 2-20, 12-2, 12-4

C

Cable
- damage, 7-2
- dynamics, 7-23
- flexing length determination, 2-7, 2-45
- load prediction reconditioning, 2-57, 7-8
- services, 2-55
- splicing, 1-15, 2-57, 2-87

Cabling, 2-23
Centrifuging, 11-19
Chain, 12-12
Circuits, hydraulics, 11-5
Cleanliness, hydraulic, 11-7
Coatings, wire
- lubricants, 1-5, 2-40,3-20
- galvanizing, 1-4, 3-12
- plastic, 1-5

Compensator, fleet angle, 10-6
Compressed sleeves, 6-19
Conductors, 2-48, 2-65
Conductor resistance, 2-39
Construction, E.M. cable, 2-4
Construction spec., E.M., 2-53
Construction spec., wire, 1-24, 5-4
Corrosion, 2-8, 3-4, 3-5
Corrosion protection, 2-18, 3-3, 3-6
Crossover spooling, 10-6
Crushing, cable, 2-9, 2-17

D

Design, equipment, 7-25
Diameter, vs. break, E.M., 2-29
Diamond screw, 10-14
Diesel hydraulics, 9-11
Documentation systems, 5-3, 5-4
Double drum efficiency, 9-19
Double drum winch, 9-1
Drag forces, 7-8, 8-11
Drum charts, 8-7
Drum construction, 10-7
Drum core, 10-12
Drum design, 9-13
Dynamic loading, 7-4

E

Electric resistance, 2-67
Electric winch drives, 8-18, 9-7, 9-8
Electro-hydraulics, 9-8
Electro-mechanical cable, 2-1
- construction, 2-4
- data, 12-4, 12-5
- diameter vs. break, 2-92
- elongation, 2-20
- manufacturing, 2-22
- physical properties, 2-85
- terminations, 6-28
- water block, 2-15

Electroline termination, 6-22, 6-28
Epoxy termination, 6-15, 6-36
Equipment design, 7-25
Equipment handling, 7-27

F

Failure reports, 5-12
Fairlead drives, 8-20
Fault location, 2-44, 2-57
Fiege fitting, 6-22
Field dressings, 3-11
Filter selection, 11-21

Fish bite, 2-9
Fleet angles, 1-23, 10-13
Fleet angle compensator, 10-16, 10-20
Fluids, hydraulic, 11-10
  filtration, 11-20
  fire resistance, 11-14
  flushing, 11-25
  synthetics, 11-14
  troubleshooting, 11-29
  water glycol, 11-15
Formulas, 12-16, 12-17

G

Galvanizing, 1-4, 3-12
Gear ranges, 8-13
Gravity settling, 11-8
Grooved drum sleeve, 2-30, 8-20, 9-13, 10-18
Grooves, sheaves, 1-21, 2-30, 4-15
Grooving pattern, 10-3

H

Hardware, marine, 12-13, 12-14, 12-15
Helically wound terminations, 6-37
Helix height, 2-11
Hydraulics, 11-1
  cleanliness, 11-17
  maintenance, 11-23
  operation, 11-3
  winch drives, 8-16, 9-8
Hydrodynamic resistance, 7-8

I

Identification, wire, 5-3
Immersed weight, static, 7-8
Inertia loads, 7-13
Inspection, cable, 2-34, 2-35
Installation, wire, 1-10
Instrumentation, winch, 8-22
Insulation, 2-22, 2-50-2-71

J

Jacketing, plastic, 1-5, 2-23

K

Kinking, wire, 2-9, 4-5

L

Lay,
  angle, 2-10
  direction, 2-10
  length, 2-12
LeBus shells, 10-3
  helix consideration, 10-3
  lagging, 1-6
  materials, 10-11
Length determination, cable, 2-45, 2-68, 2-79, 2-80
Level wind, 10-12, 10-14
Log books, 2-34, 5-11
Lubrication, 2-40, 3-2
Lubricants, 3-1
  field application, 3-12
  techniques, 3-13, 3-14

M

Magnetic marking, 2-58
Maintenance, hydraulic, 11-23
Manila line, 12-9
Material,
  cable, 2-48
  wire, 1-3
Modulus, of elasticity, 1-4, 12-3

N

Nash-Tuck splice, 1-15
Non-destructive testing, 1-30, 1-32, 2-48
Nylon line, 12-6

O

Open conductor, 2-43
Open loop circuit, 11-5

P

Performance spec., E.M., 2-54
Physical properties, E.M., 2-85
Pitch diameter, 2-12
Plastic jacketing, 1-5
Polyester line, 12-7
Polypropelene line, 12-8
Poured sockets, 6-11
Predictive cable loads, 7-8
Preform, 2-11
Preservatives, wire, 3-16
Prestressing, 2-24
Properties,
  conductors, 2-65

Properties,
plow steel, 2-72, 2-73

R

Reconditioning, cable, 2-57
Records, 2-34, 5-7, 5-9
Relubrication, wire, 3-2
Reports, failure, 5-12
Re-reeling, 2-44, 2-78
Resistance,
conductor, 2-39
hydrodynamic, 7-8
Retirement criteria
wire rope, 1-20
E.M. cable, 2-45
Rotation, 1-4
Rust preventatives, 3-8, 3-10

S

Samson line, 2-10, 2-11
Services, cable, 2-55
Serving, E.M., 2-23
Sheave,
bearing pressure, 2-69
diameter, 1-21, 2-30, 4-7, 4-15
grooves, 1-21
Shielding, 2-50
Short location, 2-93
Single drum winch, 8-1
Smooth-faced drum, 1-6, 2-29,
Snap load, 7-7, 7-19
Specifications,
cable, 2-53, 2-54, 2-74 2-75, 5-4
wire, 1-24, 5-4
LeBus shell, 10-18
Splicing,
wire rope, 1-15
E.M. cable, 2-57, 2-87, 2-91
Spooling, 1-6, 2-28, 2-29, 2-30, 2-57, 10-6, 10-18
Static load, immersed, 7-8
Steady state tension, 7-15
Storage,
wire, 2-27, 3-16
hydraulic, 11-17
Stranding, cable, 2-22
Stress relieving, wire, 1-5
Synthetic fluids, 11-14

T

Tensile loads, 7-3
Tension,
cycling, 2-8
elongation, 2-62
steady state, 7-15
Temperature correction, 2-67
Terminal,
efficiency, 6-3
velocity, 7-12
Terminations, 6-1
corrosion potential, 6-3
selection, 6-3
Terminations, installation of
mechanical, 6-22
Nicopress, 6-19
rope clips, 6-5
spelter sockets, 6-11
swaged, 6-21
thimbles, 6-9
wedge, 6-10
Testing,
E.M. cable, 2-22, 2-31, 2-48
wire rope, 1-2, 1-25, 3-8
Torque,
balance, 1-3, 2-16, 2-63
cable, 2-81
Traction winch,
efficiency, 9-19
power, 9-6
theory, 9-2
sizing, 9-11
Tread diameters, 1-21, 2-30, 9-4
Troubleshooting, hydraulic, 11-29
Twist balance, 2-17

V

Virtual mass, 7-13
Viscosity, class, 11-15

W

Water,
blocked core, 2-15
glycol fluid, 11-15
Wedge terminations, 6-10
Winch,
documentation, 5-20
single drum, 8-1
capacities, 8-7
construction, 8-24

Winch,
  single drum,
    controls, 8-19
    instrumentation, 8-22
    ordering, 8-26
    power options, 8-15
    theory, 8-2
  sizing, 8-26, 9-11
  traction, 9-1
Wire rope,
  breaking load, 1-2
  coatings, 1-4, 1-5
  construction, 1-2
  data, 12-2, 12-3
  modulus of elasticity, 1-4
  offset yield 2%, 1-4
  preservation, 3-16
  rotation, 1-4
  specification, 1-24
  stress relieving, 1-5

Z

Zero loads, 7-6

UNCLASSIFIED, 9/82

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE

READ INSTRUCTIONS BEFORE COMPLETING FORM

1. REPORT NUMBER

2. GOVT ACCESSION NO. A132 513

3. RECIPIENT'S CATALOG NUMBER

4. TITLE (and Subtitle)

Handbook of Oceanographic Winch, Wire, and Cable Technology.

5. TYPE OF REPORT & PERIOD COVERED

Technical

6. PERFORMING ORG. REPORT NUMBER

7. AUTHOR(s)

Alan H. Driscoll, Editor

8. CONTRACT OR GRANT NUMBER(s)

N00014-82-G-0116

9. PERFORMING ORGANIZATION NAME AND ADDRESS

Graduate School of Oceanography
University of Rhode Island
Narragansett, RI 02882

10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS

11. CONTROLLING OFFICE NAME AND ADDRESS

Office of Naval Research DET
NSTL Station
Mississippi 39529

12. REPORT DATE

18 October 1982

13. NUMBER OF PAGES

432

14. MONITORING AGENCY NAME & ADDRESS(if different from Controlling Office)

15. SECURITY CLASS. (of this report)

UNCLASSIFIED

15a. DECLASSIFICATION/DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release - distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

Report prepared jointly by Office of Naval Research and the National Science Foundation.

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Oceanographic equipment, deep sea winches, wire ropes, electro-mechanical cables, techniques for standard use, equipment design, hydraulic systems.

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This handbook contains specific data relative to the technology involved in the selection, operation, and maintenance of oceanographic winch, wire, and cable systems. Particular attention is given to the development of a comprehensive guide.

DD FORM 1473 1 JAN 73 EDITION OF 1 NOV 65 IS OBSOLETE
S/N 0102-014-6601

UNCLASSIFIED, 9/82

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

For the user, specific details are given concerning the development of an approach which will increase the longevity of oceanographic equipment currently in use and which can be applied to new acquisitions. The design of lowered packages to increase their hydrodvnamic characteristics and reduce drag are discussed, along with a careful survey of various styles of wire rope and electro-mechanical cable terminations. This handbook is designed for use by scientists, engineers, ships' personnel, and technicians who are required to use oceanographic winch systems.

END

FILMED

10-83

DTIC